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OPTIMIZATION OF CENTRIFUGAL PUMPS OPERATION FOR LEAST COST AND MAXIMUM AVAILABILITY

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ABSTRACT

There are five M.O.L. Centrifugal pumps "Main Oil Line" in Ghani oil field for transporting crude oil from Ghani oil field to Joffra oil field at a distance of about 81.25 miles (130Km) then from Joffra oil field to Ras Lanuf Terminal at a distance of about 84.40 miles (135Km).

The five centrifugal pumps do not all have the same specifications. Three of these pumps have the same power (255 Kw) and are working with five impellers but the other two pumps have the same power (150 Kw) and are working with six impellers. The efficiency and the head are the same for each pump. So the total power in the pumping system is (1,065 Kw). Usually two out of five centrifugal pumps are operational the other three pumps are on standby in case of failure of one or more pumps. These pumps are 24 years old and subjected to routine maintenance.

This project presents a study to optimize the existing pumping system in order to make the capital cost less. The main change will be in pumping station design mainly to decrease the number of the existing centrifugal pumps with less cost and to meet Veba Oil Operations (VOO) requirements. As mentioned, the system is required to transport 45,000 BPD (298 m^3/h), during the expected life of the project, which is through the pumping system from Ghani oil field to Joffra oil field then to Ras Lanuf Terminal.
This study offers different pump design to evaluate and select the optimum design according to the lowest cost, highest reliability and availability.

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Chapter 1

INTRODATION

1.1. Background

Veba Oil Operations (VOO) is one of the biggest companies in Libya. It operates five oil fields Joffra, Tibisti, Amal, Ennaga, and Ghani fields and one Terminal Ras Lanuf. Figure (1.2) on page 3 shows all the oil fields and the pipeline network for the Ras Lanuf Terminal.

The Ghani oil field is situated in the south of Libya. It was discovered in 1978 at a depth of 1.83 Km.

The production of this field is about 45,000 *BPD* (298 m^3 /hr). The field comprises three separate reservoirs. More than one hundred wells have been drilled in this field. Crude oil is piped from Ghani oil field to the Ras Lanuf Terminal over a total distance of about 165.6 miles (265 Km) passing through Joffra and Assida Junction. Figure (1.1) shows the crude oil pipeline from Ghani oil field to Joffra oil field then to Ras Lanuf Terminal.

Ghani crude oil is classified as a paraffin crude oil having a gravity of 60 ^{o}F, (0.823) *API*, and crude oil density of 40.2 *API*. Ghani crude oil has kinematics viscosity 6.7 cSt, (8.1×10^{-6} m^2/sec) at 100 F^0, (37.78 ^{o}C).

Three methods have been used by the Veba Oil Company in all oil fields to get the crude oil from the wells. Natural flow comes first for a certain time. When the flow rate of crude oil starts decreasing, electrical submersible pumps are used to start extracting oil from the wells. Jack pumps are the last method of extracting oil.

The Ghani oil field includes five stations, Ghani station is the main, Facha station, Tagrifet station, Zenad station and Eddib station. Both Ghani station and Zenad station have oil / gas / water separation facilities. The oil goes from Facha and Tagrifet stations to Zenad station then directly to the main station (Ghani station). Crude oil is transported from Ghani oil field to Joffra oil field over a distance of about 81.25 miles (130Km) then from Jofra oil field to Ras Lanuf Terminal over a distance of about 84.4 miles (135Km).

Figure (1.1) Crude Oil Pipeline from Ghani oil field to Joffra oil field then to Ras Lanuf Terminal.

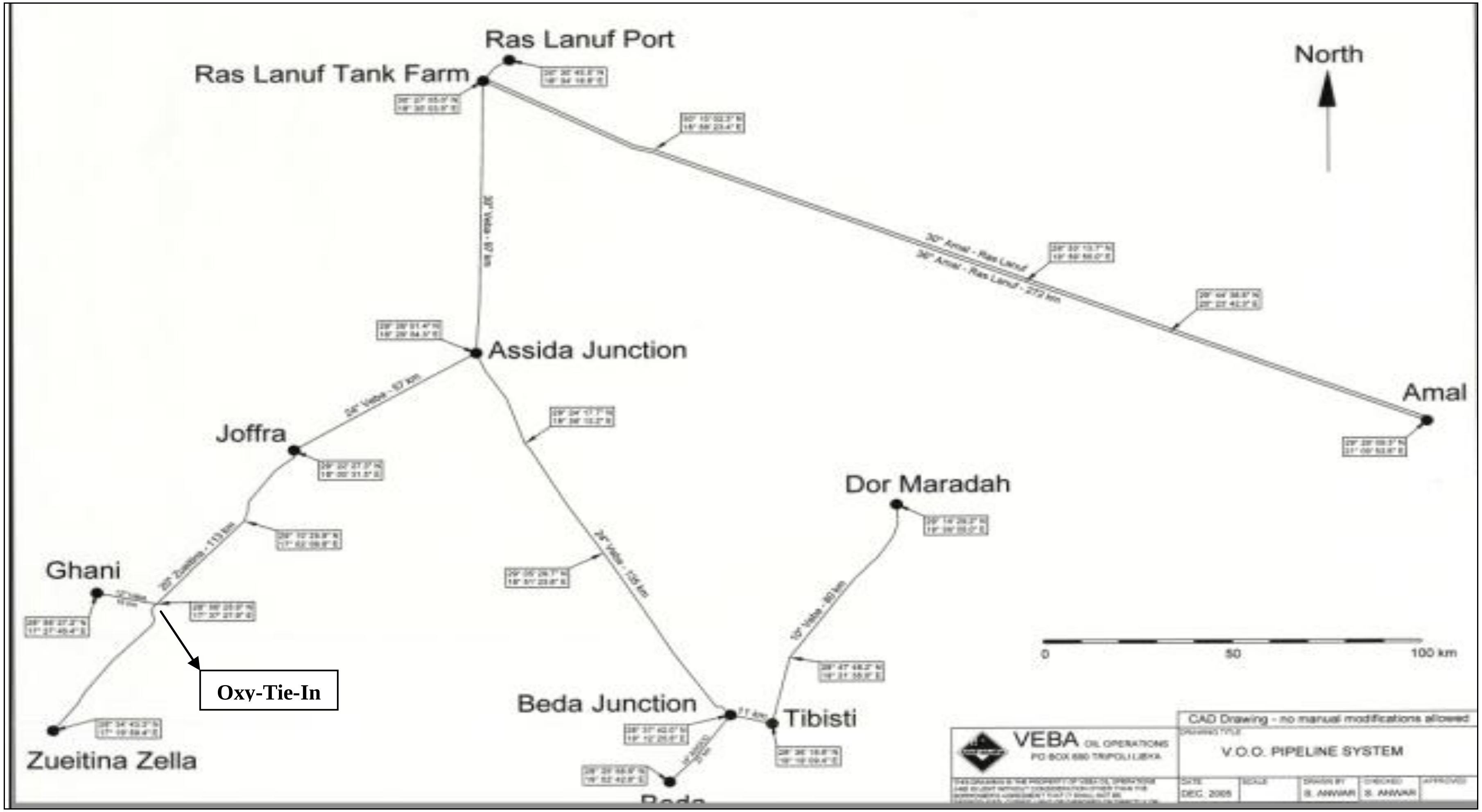


Figure (1.2) The Pipeline Network for the Ras Lanuf Terminal

1.2. Project Objectives

This study attempts to arrive at the optimum pumping system at the lowest capital and annual cost. The data used in this study is real data taken from Veba Oil Operations (Ghani oil field). Therefore the main objectives of this project can be summarized as:

- To collect the data dealing with the system.
- To analyse the hydraulic design.
- To study the pumping system requirements.
- To select the pumping systems.
- To optimize the requirements of the pumping system.
- To study the system cost and sensitivity of pumping system.
- To study the reliability of the system.
- To study the availability of the pumping system.

Chapter 2

PREVIOUS WORK

- Centrifugal pump operation involves more than simply switching the pump on and directing the discharge flow to the required delivery point. A few very essential operating guidelines must be adhered to early. For example, care must be taken to cater for potential bearing or seal failure, premature erosion of internal wetted surfaces, and metal- to –metal contact of internal rotating and stationary surfaces.

The most common causes of centrifugal pumps failure, separate from those causes associated with maintenance and/or design but associated with how the pumps are operated, may be summarized as following:

1. Insufficient suction pressure to avoid cavitation.
2. Prolonged operation at lower than acceptable flow rate.
3. Improper operation of pumps in parallel.
4. Excessively high flow rate for net positive suction head available ($NPSH_A$).[26]

- Pump and motor selection, properly selecting a pump and motor is very important to meet design requirements. However, not only is the capacity and pressure of concern, other parameters such as motor horsepower, efficiency single or parallel operation, shut off head and variable speed operation are among other important points to consider. The reliability team prefers lower speed equipment such as 1180 and 1750 rpm equipment.

Parallel pumping is an important design choice since it allows the system to continue to operate one pump while the primary pump is off line for maintenance.[24]

- There is a 34 inch pipeline system transporting crude oil from the Sarir field to Tobruk Terminal this is 320 miles (512 km) away. The system is 35 years old and no longer efficient. Accordingly, this study is focused on the components that would have a major impact on the total cost of the system.

These components are the pipe itself (pipe diameter, wall thickness and pipe material and grade). The pumping system required transporting the required capacities of the oil and the other components that will affect directly or indirectly the performance and the total cost of the system should also be taken into consideration.

This study offers different pipeline designs to evaluate so as to be able to select the optimum design according to the lowest total cost (capital and annual cost) and highest reliability.

The main conclusions drawn from this study of transporting the above capacities of crude oil from Sarir to Tobruk with lowest capital and annual operating cost are summarized as follows:

A 24 inch pipe diameter with 0.312 inch, (7.92mm) wall thickness and grade *API 5L X 46* would be the optimum pipe size. This size of pipe requires three pumping stations along the pipeline, the first at the beginning, the second at 120 miles (192 km) and the third at 225 miles (360 km). Three heating stations would also be required, one at each pumping station to deal with the fact that the crude is waxy.[30]

- Further improvement of centrifugal pumps in order to increase the efficiency and head coefficient of the impeller is an important problem whose solution will make it possible to reduce the consumption of electrical energy. The inclination of impeller vanes should be increased to a final value of 80-130 degree near the outlet on a length corresponding to 15-50 % of the total vane length. To optimize the flow regimes it is expedient to install in the zone of the increased vane inclination 1-5 additional short vanes with a length amounting 0.6-1.2 of the length of the swept-forward tips of the main vanes. Impellers with an increased inclination of the outlet tips of the vanes and with additional short vanes in this zone increase the pump head by 30-95 % for the same dimensions of the impeller and its rotative speed, increase the efficiency by 5-15

% at high deliveries and by 1-3 per cent at rated deliveries and also increase the delivery by 5-80 %. [35]

Chapter 3

PIPELINE SYSTEMS

3.1. Introduction

An oil piping system introduces those pipelines that transport oil and petroleum products from production sources and tank fields to storage facilities and final user locations.

In the design of a pipeline for the transportation of oil, it is very important to consider many aspects of pipeline design as well as project economics in determining the optimum pipeline system to transport a commodity from a source to destination on a technical or engineering level.

The hydraulic design is the process of considering the physical characteristics of the product to be transported, the quantities to be transported, and the pipeline route and topography and the values of pressures. Identifying the pump station with referring to the hydraulic characteristics of the system is also part of the hydraulic design. There may be many different hydraulic designs for any given pipeline design configuration and route.

For any hydraulic design there are several mechanical system designs that can be developed to meet the criteria of the design basis and deliver the product from refineries to the destination. For many reasons, design principles and the materials used differ considerably within the pipeline industry, according to the fluid transported. For example, oil, gas and water need to be transported over long distances, and this can be achieved economically only by the use of a high-pressure pipeline. Furthermore, oil and gas are valuable products so leakage from pipelines is unacceptable for commercial, environmental and safety reasons.

3.2. Hydraulic Design

The hydraulic design integrates the physical characteristics of the transported commodity along a given pipeline route, within specified operating conditions as established in the design basis. The result of the hydraulic design is identification of the total system energy required to meet the design criteria. In addition, the hydraulic calculations indicate a range of feasible pipe diameters and preliminary spacing of pump stations along the route.

When the design is finalized (i.e. the route selected, pipe line size determined, and type of pipe selected), the hydraulic calculations are refined to determine the conditions for overpressure control during line shut-off and surges during operation. Hydraulic calculations can also be made for the variables in the operating conditions (temperature, ranges of viscosities for products pipeline, etc.) and for future expansion of system capacity.[15]

3.2.1. System Energy

System energy, which means power, required moving the crude oil through the pipe. It requires calculating the total pressure drop in the pipeline system.

Total pressure drop in a pipeline system comprises three components:

1. Static pressure drop, due to changes in elevation.
2. Friction pressure drop, due to flow rate, fluid properties, and pipe characteristics.
3. Velocity head.

The velocity head is one of the elements making up the total dynamic head, but the value is small so that it can be disregarded. The velocity head in the pipe is the strength of the stream at the outlet of the open pipe. This head should also be considered in accurate testing, since it is part of the total head and will therefore affect the duty of the pump. Friction head losses, on the other hand, are the dominant effect in most liquid pipeline systems and can be calculated by using the Darcy-Weisbach formula.

Friction head losses $$H_f = f\,\frac{L \times V^2}{D \times 2 \times g} \qquad (3.1)$$

Where:

f = the friction factor which is can be determined from an equation developed by Swamee & Jain formula, (equation 3.2).[22]

$$f = \frac{0.25}{\left[\log\left(\frac{K}{3.7D} + \frac{5.74}{\mathrm{Re}^{0.9}}\right)\right]^2} \qquad (3.2)$$

Or by using a Moody chart, figure (3.1) on page 12.

Where:

H_f = Friction Head losses (m)

L = Pipe length (m)

V = Average pipe velocity = $\frac{Q}{A} = \frac{Q \times 4}{\pi \times D^2}$ (m/sec)

Q = Flow rate, (m^3 /sec)

D_I = Inner diameter of pipe (m)

g = Gravitationnel constant (9.81m /sec^2)

K = Absolute roughness (m), table (3.1) shows the typical values for different type of pipe

R_e = Reynolds number $$R_e = \frac{VD}{v} \qquad (3.3)$$

v = Kinematics viscosity, (m^2 /sec)

Head (*H)* and pressure (*P*) are used in system energy; the conversion from head to pressure is given by:

$$\text{Liquid Head in psi} = \frac{head(feet) \times sg}{2.31} \qquad (3.4)$$

Type of pipe(new, clean, condition)	Absolute Roughness meter
Draw tubing – glass, brass, plastic	1.52×10^{-6}
Commercial steel or wrought iron	4.52×10^{-5}
Cost iron – asphalt dipped	1.21×10^{-4}
Galvanized iron	1.52×10^{-4}
Cast iron - unquoted	2.59×10^{-4}
Wood stave	$1.8 \times 10^{-5} – 9.1 \times 10^{-5}$
Concrete	$3.0 \times 10^{-4} – 3.0 \times 10^{-3}$
Riveted	$9.1 \times 10^{-4} – 9.1 \times 10^{-3}$

Table (3.1) Roughness Value for Different Pipes [19]

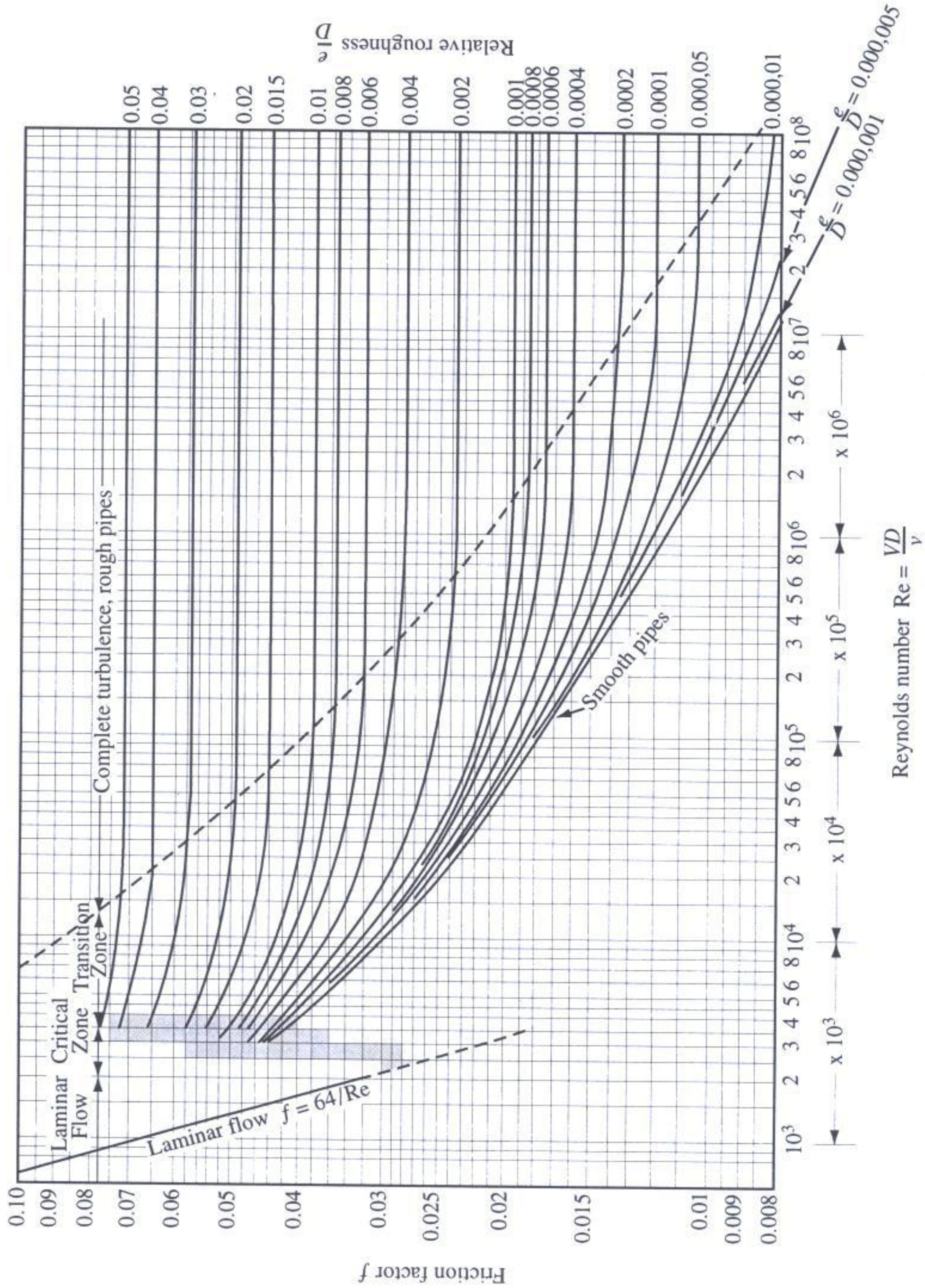


Figure (3.1) Moody Diagram [14]

The inside diameter (*D*) preliminary choice depends on the range of operating pressures. Basically, the velocity range is between 0.60 to 2.13 m/sec from the past experience table (3.2) and figure (3.2) shows the mean values of velocities for different pipelines below.

The fluid must be kept below maximum velocity (2.13 m/sec), to prevent such problems as erosion, noise and water hammer and the fluid must also be kept above minimum velocity 0.60 m/sec, to minimize surging and to avoid deposition of solids.

Pipeline	Velocity m / sec
Local water service pipe systems	0.48 – 0.79
Water main and pipelines	0.91 – 3.04
High pressure gas lines	4.99 – 15.24
Oil pipelines	0.60 – 2.13
Compressed air lines	15.24 – 30.48
Low pressure gas line	4.99 – 8.07

Table (3.2) Mean Values of Velocity for Different Pipelines [19]

By using the equation (3.1) to calculate the friction head losses at different diameters and wall thicknesses, table (3.3) shows different results for Velocity, Reynolds Number, Friction factor and Friction head losses appropriate to different pipe diameters.

Diameter D		Velocity V	Reynolds No, R_e	Friction Factor f	Friction Head losses H_f
inch	M	m/sec			m
12 (Ghani-Oxy-Tie-In)	0.305	1.5	56481.5	0.0208	132
20 (Oxy Tie-In-Joffra)	0.508	0.677	42459	0.022	114

Table (3.3) Pipeline Materials Geometry and Properties

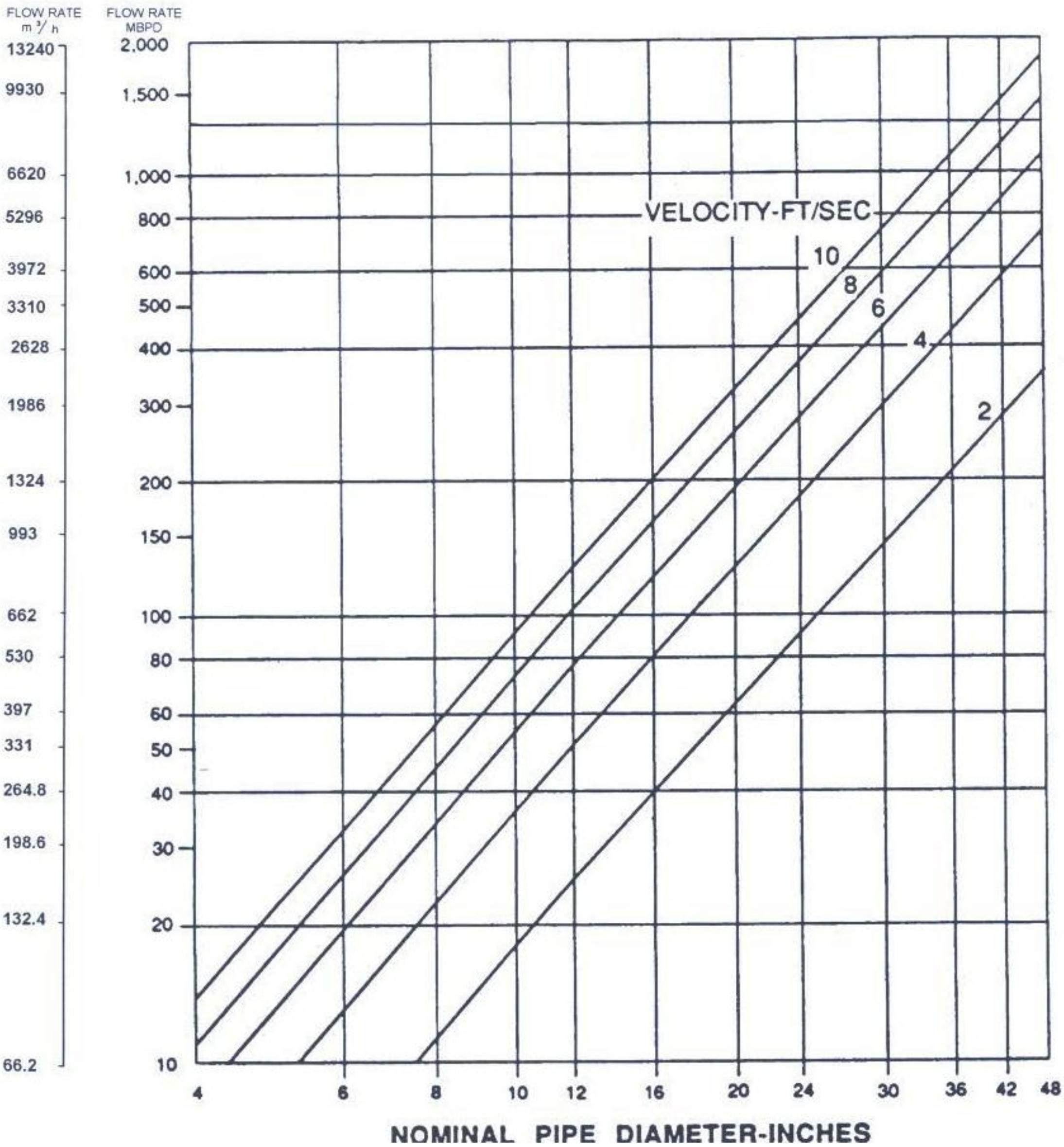


Figure (3.2) Flow Rate versus Nominal Pipe Diameter for Recommended Viscosities for Crude Oil Pipe Line [15]

Pipe diameters will not be acceptable due to their velocities being out of an acceptable range for oil pipelines, 0.60 to 2.13 m/sec to ensure friction losses are minimized.
Minor losses at valves and fittings installed on the line are normally ignored for long pipeline systems.

3.2.2. Hydraulic Gradient

The hydraulic gradient is a profile representing the static head at any point in the pipeline system relative to a common datum elevation. The latter is usually mean sea level. The ground elevation is represented by the route elevation profile to the same datum. The energy added to the system though a pumping station is plotted above the elevation profile. Head losses due to friction, etc., are also shown graphically. For a pipeline system with constant parameters along the system, such as viscosity, specific gravity and diameter, the hydraulic gradient will be a straight line with slope equal to the friction loss per unit of length (H_f) for a specific flow rate. Therefore, the actual pressure in the pipeline at any point along the route is the difference between the hydraulic gradient and the ground elevation.

The hydraulic gradient is used for identifying hydraulic control points on a pipeline route. A hydraulic control point is a high elevation point that will govern the inlet head for a section of pipeline. The station discharge head is usually selected on the basis of the allowable pressure rating for valves and fittings or the Maximum Allowable Operating Pressure (*MAOP*) of the selected pipe. [15]

3.2.3. Maximum Allowable Operating Pressure (MAOP)

The *MAOP* is the limit of internal pressure allowed for straight pipe by section defined by 404.1.2 of code.[15]

$$MAOP = 2 \times \mathrm{S} \times \frac{t}{D} \times F \times E \times T \qquad (3.5)$$

Where:

S = Specified minimum yield strength of pipe *SMYS*, psi, (*MPa*) table (3.4)

D_O = Outer diameter of pipe, (m)

t = Wall thickness of pipe, where "t" can not be less than that shown in the table (3.5)

F = Construction type design factor, showing in table (3.6)

E = Longitudinal joint factor, showing in table (3.7)

T = Temperature derating factor, showing in table (3.8)

Pipe Grade	Specified Minimum Yield Strength (SMYS)		Ultimate Tensile Strength, Minimum	
	PSI	MPa	PSI	MPa
A25	25,000	172	45,000	310
A	30,000	207	480,000	331
B	35000	241	60,000	413
X42	42,000	289	60,000	413
X46	46,000	317	63,000	434
X52	52,000	358	66,000	455
X56	56,000	386	71,000	489
X60	60,000	413	75,000	517
X65	6,5000	448	77,000	530
X70	70,000	482	82,000	565
X80	80,000	551	90,000	620

Table (3.4) Specified Minimum Yield Strength for Different Grades [12, 21]

LEAST NOMINAL WALL THICKNESS [1]

Least Nominal Wall Thickness (Inches)								
		Plain End Pipe[A]				Threaded Pipe	All Pipe	
Nominal Pipe Size (Inches)	Outside Diameter (Inches)	Class I Location	Fabricated Assemblies Class I Locations	Class 2 Locations	Class 3 & 4 Locations	All Class Locations	Compressor Stations	
1/8	0.405	.035	.065	.065	.065	.068	.095	Plain end or threaded
1/4	0.540	.037	.065	.065	.065	.088	.119	
3/8	0.675	.041	.065	.065	.065	.091	.126	
1/2	0.840	.046	.065	.065	.065	.109	.147	
3/4	1.050	.048	.065	.065	.065	.113	.154	
1	1.315	.053	.065	.065	.065	.133	.179	
1-1/4	1.660	.061	.065	.065	.065	.140	.191	
1-1/2	1.900	.065	.065	.065	.065	.145	.200	
2	2.375	.075	.075	.075	.075	.154	.218	
2-1/2	2.875	.083	.085	.085	.085	.203	.203	
3	3.500	.083	.098	.098	.098	.216	.216	
3-1/2	4.000	.083	.108	.108	.108	.226	.226	
4	4.500	.083	.116	.116	.116	.237	.237	
5	5.563	.083	.125	.125	.125	.258	.250	Plain end only
6	6.625	.083	.134	.134	.156	.280	.250	
8	8.625	.104	.134	.134	.172	.322	.250	
10	10.750	.104	.164	.164	.188		.250	
12	12.750	.104	.164	.164	.203		.250	
14	14.000	.134	.164	.164	.210		.250	
16	16.000	.134	.164	.164	.219		.250	
18	18.000	.134	.188	.188	.250		.250	
20	20.000	.134	.188	.188	.250		.250	
22,24,26	22,24,26	.164	.188	.188	.250		.250	
28,30	28,30	.164	.250	.250	.281		.281	
32,34,36	32,34,36	.218	.250	.250	.312		.312	
38,40,42	38,40,42	.250	.312	.312	.375		.375	

[A] For tubing in wall thickness over 0.035 inches, the wall thickness may be obtained by interpolating between the pipe outside diameters listed above. Instrument, control and sample piping are not limited by table.

[1]From A.G.A. GPTC Guide For Gas Transmission and Distribution Piping Systems, Section 192.103 and Table 192.103i.

Table (3.5) Least Nominal Wall Thickness for Different Pipe Diameters [16]

The design factor to be used in the design formula is determined in accordance with table (3.6).

Class Location	Design Factor, *F*
1	0.72
2	0.60
3	0.50
4	0.40

Table (3.6) Construction Type Design Factor [16]

Where:

1. Class 1 locations include wastelands, deserts, rugged mountains, farmland and similar areas.

2. Class 2 locations include fringe areas around cities or towns and farm or industrial areas with a specified population density. This class is between class 1 and 3 locations.

3. Class 3 locations include areas subdivided for residential or commercial purposes with specified building density of specified type.

4. Class 4 locations include areas where multi-storey buildings are prevalent, where traffic is heavy or dense, or where there is numerous other underground utilities. [18]

The construction type design factors (*F*) are also specified in pipeline codes for that cross or runs parallel to roads and railroads.
The effect of the construction type factor is to lower the allowable operating pressure for a given size, weight, and grade of pipe.[18]

The longitudinal joint factor (*E*) to be used in the design formula is determined in accordance with table (3.7).

Specification	Pipe Class	Longitudinal Joint Factor, *E*
API 5L	Seamless	1.00
	Eclectic resistance welded	1.00
	Eclectic flash welded	1.00
	Submerged arc welded	1.00
	Furnace butt weld	0.60

Table (3.7) Longitudinal Joint Factor for Different Pipes [16]

The temperature derating factor (*T*) to be used in the design formula is determined in accordance with the table below. The temperature derating factor for steel pipe varies from one for operating temperatures of 250 ^{0}F (121 ^{0}C) or less to 0.867 for an operating temperature of 450 ^{0}F (232 ^{0}C).

There are other specifications and limits that may apply to pipeline pressure. Limits on operating pressure may be set by the fluid being transported.[18]

Fluid Temperature		Temperature Derating Factor, *T*
^{0}F	^{0}C	
250 or less	121.1	1.000
300	148.9	0.967
350	176.7	0.933
400	204.4	0.900
450	232.2	0.867

Table (3.8) Temperature Derating Factor at Different Temperatures [16]

Wall thickness (*t*) for the calculation of the *MAOP* excludes additional thickness for corrosion allowance or imposed stresses such as concentrated loads at supports, thermal expansion or contraction and bending to maximize the strength of the pipe.

With different wall thicknesses, grades are based on experience in the calculation of the maximum operating pressure for each wall thickness and grade.

$$MAOH\ (\mathrm{m}) = \frac{9.807 \times MAOP(KPa)}{Sg} \tag{3.6}$$

$$MAOH\ (\mathrm{feet}) = \frac{2.31 \times MAOP(Psi)}{Sg} \tag{3.7}$$

$$\text{Total system head } (H_T) = H_f + h_{St} \tag{3.8}$$

Where:

H_f = Pipeline friction, Psi (KPa)

h_{St} = Static head, (m)

The elevation head at Ghani oil field is lower than the elevation head at Joffra oil field by 40 m. Table (3.9) shows the *MAOP* for Ghani oil field to Joffra. The wall thicknesses shown in Table (3.9) below are chosen according to equation (3.5).

API 5L X 52

Wall Thickness (inch)	0.406 (Ghani-Oxy-Tie-In)	0.312 (Oxy-Tin-In-Joffra)	Total
System Head Losses, H_f	132 m	114 m	246 m
Total Pressure Head	1062 KPa	919 KPa	1981 KPa
Maximum Allowable Operating Head (*MAOH*)	207.34 m (1672 KPa)	97 m (782 KPa)	304.34 m (2454 KPa)
Maximum Allowable Operating Pressure (*MAOP*)	17×10^3 KPa	8.12×10^3 KPa	25.12×10^3 KPa

Table (3.9) *MAOP* for Ghani oil field to Joffra oil field

3.3. Mechanical Design

Mechanical design of a pipeline system is the selection of materials, including type of steel, diameter and wall thickness of pipe, as methods of support and/or restraint for the system in response to the loadings and stresses imposed on the pipeline system by physical pressures and forces such as the internal and external design pressures; static loadings and weight effects of the pipe, fluid, and soil; dynamic loadings from wind, waves, earthquake, etc.; and relative motion of connected components. These factors impose loadings on the pipe and result in longitudinal, hoop, and radial stresses which must be evaluated in the mechanical design of the piping system.[15]

In addition to the mechanical factors that affect the allowable stress levels for design, the grade of steel and wall thickness determine welding procedures and affect construction cost.

The mechanical design of the pipeline, with respect to restraint against longitudinal, or axial and radial motions, considers the pipeline as a unit and must provide that sufficient flexibility is designed into the system to ensure that expansion or contraction as a result of the internal or external loadings does not cause excessive stresses in the piping material, bending moments at joints, or excessive forces or moments at points of connection to equipment or at supports. Line pipe is manufactured by several methods, the most common being seamless, electric resistance welding (*ERW*) and submerged arc welded (*SAW*) in the form of longitudinal and helical (spiral) welds. Each has advantages and disadvantages for different uses and there are also economic and availability considerations.

With respect to the mechanical design of a pipeline, the characteristic of a line pipe which is of critical interest is the specified minimum yield strength (*SMYS*) of the material. *API 5L* specification for line pipe is available in various strength grades ranging from Grade *B*, rated at 35,000 psi (241 MPa) to *X* 80, where the number refers to the *SMYS* in kips per square inch (ksi), a kip being 1000 Ib.

There is some advantage to the higher strength grades, principally in that wall thickness may be reduced. In some cases this may have an economic impact on the project, since thinner walls translate into lower steel tonnage for the entire pipeline system, and this may be a significant factor, even though higher grades of pipe cost more per ton.

Cost saving can also result from reduced time required to field weld the thinner wall section. There are other considerations, which will affect the decision to higher strength, thinner wall pipe. These include aspects of construction which are the result of experience in the field, such as the way the pipe handles with regard to field bending, laying stresses, tendency to go “out of round” etc. In addition, there may be limitations placed on the grade of pipe and wall thickness used for a particular project, particularly for a system which will be in “sour” or corrosive service.

3.3.1. Pipe Diameter

In the hydraulic design of a pipeline system, line size is initially based on a preliminary choice of diameter and wall thickness from experience and from simplified charts. Further calculations are needed to verify the selection and finalize the system design based on the code requirements as wall as considerations for project cost and material availability.

For most pipeline systems, the pipe cost, which is a function of the diameter and wall thickness, will be the highest material cost in the system. In addition, the size of pipe will have a direct effect on the cost of installation. Therefore, total project cost is affected by the selection of pipe size. For this reason, it is important to optimize the pipe diameter, wall thickness, and grade of steel to be used so that the overall project cost is contained.
As discussed in the hydraulic design section of this chapter, the diameter of pipe is a function of the design flow rate, and mechanical design considerations have little effect on diameter selection. However, internal and external pressure, allowable stress, etc., do affect the final design of the wall thickness for the selected diameter.

3.3.2. Pipe Material

Steel pipes normally used for oil and gas pipelines are those complying with *API* standards *5L* or *5LX*. Standard *5L* includes two grades of steel, grades *A* and *B* while standard *5LX* covers eight grades: *X42, X46, X52, X56, X60, X65, X70* and *X80* the number representing in thousands of pounds per square inch, with the specified minimum yield strength (SMYS). These standards lay down the outside diameter and wall thickness of the pipe. The figures for nominal sizes are approximately the same as the inside diameters, and they are applied in particular to the mating components of a pipe such as fittings and bends. [36]

The outside diameter is a dimension fixed by the manufacturing tools. Variations in wall thickness affect the inside diameter only.

In the hydraulic design, a preliminary determination of wall thickness is based on experience for the preliminary selection of pipe diameter and grade of steel. The actual design of a system must reflect code requirements for the wall thickness.

3.3.3. Pipeline Pigs

A "smart" pig is an instrumented device which travels internally along a pipeline, monitoring the operating parameters (flow, temperature, etc.) and physical condition of the pipe (wall thickness, corrosion, out-of- roundness, etc.). Pigs are also used to "listen" for the acoustical traces of leaks. Simple pigs, or spheres, are sometimes used to mark the transition between two commodities in a multiproduct pipeline.

The information the pig collects must be relayed to the master control centre and the pumping stations along the route must have facilities designed for the handling of pigs, including launching and receiving traps.[15]

Pigs play an important part in operation and maintenance work on pipelines. The pigs are moved with the oil through the underground lines cleaning, deposits of sediment and other foreign matter. Their moving force is the stream of oil in the line.

Chapter 4

PUMPS AND PUMPING SYSTEMS

4.1. Introduction

Pumps can be used to transfer liquids from one place to another by converting mechanical energy into pressure energy. In this case pressure is applied to the liquid forcing it to flow at the desired rate and to overcome friction losses in piping, fittings and valves.

To design the pumping system, the following factors should be considered as fluid properties determination of end use requirements and understanding environmental conditions. Other design aspects include inertia bases and variable speed pumps.

Pump selection must be taken into account foundation preparation, shaft alignment, and soft foot correction and vibration analysis.

That is why reliability of centrifugal pumps is of critical importance to daily operations. Continuous and reliable operation of utility systems using these types of pumps such as process of transporting oil and gas is essential to provide the utilizing areas.

It is also equally important to optimize the pumping systems layout i.e. a “clean” layout is an efficient layout.

Maintenance Activities

Regular maintenance keeps pumps operating well and allows detection of problems ahead of time to schedule repairs. Regular maintenance also shows the capacity and efficiency deterioration that can occur long before pumps fail.

There are two types of maintenance activities; predictive maintenance that concentrated on tests and inspection; and preventive maintenance such as lubrication, removal of contaminates and periodic adjustment.

4.2. Pump Types

Pumps can be classified into two fundamental types based on the manner in which they transmit energy to the pumped media. Kinetic energy pumps and positive displacement pumps.

4.2.1. Positive Displacement Pumps

Positive displacement pumps, the moving element (piston, plunger, lobe, or gear) displaces the liquid from the pump casing or cylinder and, at the same time, raises the pressure of the liquid.

In pumps of this type, energy is periodically added by application of force to one or more movable boundaries of any number of liquid containing volumes, resulting in a direct increase in pressure up to the value required to move the fluid through valves or port into the discharge line. Displacement pumps are essentially divided into reciprocating or rotor types.

4.2.1.1. Reciprocating Pumps

A reciprocating pump is a positive displacement pump that works on the principle of a reversing piston motion within a cylinder, drawing in fluid on the ingoing stroke and delivering it under pressure on the outgoing stroke. This is achieved using one way valves (or equivalent piston swept ports), on both the suction and delivery side. A variety of configurations is possible dependent on the mechanical system used to derive the reciprocating piston motion; the number and arrangement of cylinders; the valving system etc. Reciprocating pumps are used for small quantity high pressure duties where their efficiency can exceed that of a centrifugal pump. Reciprocating pumps divide into the following types.[39]

4.2.1.1.1. Plunger Pumps

A plunger pump is shown in figure (4.1) it can apply for transfer and for metering service. Such pumps are available in single and multicylinder models. The plunger contains the crosshead, driven by a camshaft arrangement. The capacity of the pump can be adjusted by changing the stroke, the rotating speed of the pump or both. The stroke of the pump is changed by the eccentric pin setting.

In the suction stroke, as the plunger retracts, the suction valve opens causing of the liquid within the cylinder. In the forward stroke, the plunger then pushes the liquid out into the discharge header.

The gland packing help to contain the pressurized fluid within the cylinder. The plungers are operated using the slider crank mechanism. Usually, two or three cylinders are placed alongside and their plungers reciprocate from the same crankshaft.

The plunger can be removed for replacement without disturbing the shaft assembly; the pumps are, therefore, well suited for metering applications. The pumps are driven by constant or variable speed motors with gear reducers. [41]

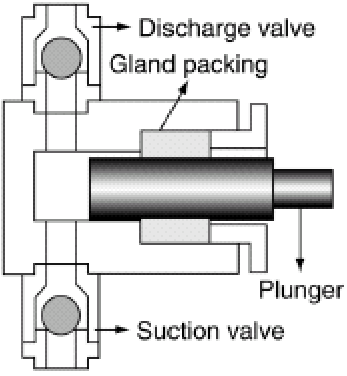


Figure (4.1) Plunger Pump

4.2.1.1.2. Diaphragm Pumps

Any product that can flow through a pipe can be transferred by means of a diaphragm pump. Liquids such as Adhesives, Asphalt, Bitumen, Emulsion, Latex, Resins and Polymers, as well as organic and inorganic chemicals and virtually any industrial liquid product including liquids with high solids content and with large particle sizes can be effectively pumped.

Diaphragm pumps have a simple operating principle. Two diaphragms fixed to both ends of a centre rod, actuated by air pressure, pump the material. As shown in figure (4.2), compressed air enters the air chamber "a" moving the centre rod to the left, forcing the material out of the material chamber "A" while the material is drawn into the material chamber "B". When the centre rod is at full stroke to the left, the air transfer switch valve changes supplying air into the air chamber "b". The centre rod then moves right, forcing out the material from the material chamber "B", at the same time, drawing more material into the material chamber "A". Continual movement of the material is chamber is achieved by repeating the operation.[37]

Infinitely variable flow control can be achieved simply by opening or closing a valve on the discharge or by increasing or decreasing the air supply.

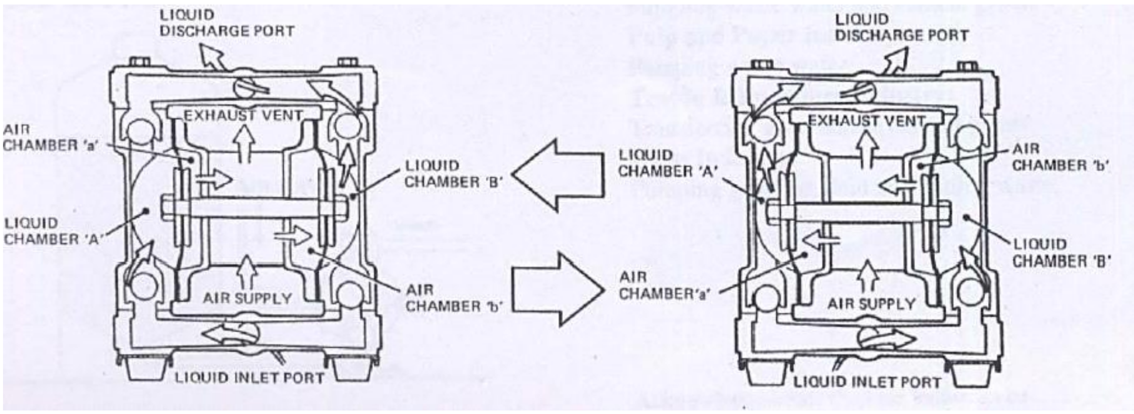


Figure (4.2) Diaphragm Pumps [37]

4.2.1.2. Rotary Pumps

Rotary pumps are also positive displacement machines but differ from reciprocating pumps in that delivery is continuous and thus smoother, although internal losses are normally somewhat higher through slip (internal leak-back). Slip increases with increasing pressure, thus rotary pumps are less suited to developing high pressures than reciprocating pumps. Compared with centrifugal pumps they are much more restricted in practical sizes and thus deliveries.

Rotary pumps are a particularly useful type since they are suitable for handling a wide variety of fluids from the very lowest to the highest viscosities (with suitable modification of design where very high fluid viscosities are concerned). However, they are suitable only for handling of clean fluids from light oils upwards, although many are equally suitable for water and some types are eminently suitable for handling gases. As well as rotary pumps divide into following types.

4.2.1.2.1. Gear Pumps

Gear pumps are also suitable for high pressure working, especially with oil fluids. This, however, places a premium on detail design, precision workmanship and calls for particular attention to reduce internal leakage to a minimum, as well as attention to bearing design to resist torque and pressure loads. Gear pumps are divided into two types.

- **External Gear Pump**

External gear pumps are the simplest in design consisting of two intermeshing gears of the same diameter and form mounted on separate spindles, one gear shaft being driven whilst the other idles, as shown in figure (4.3). This type is readily produced in a wide variety of sizes.

Pumping action is produced by nature of the fact that during rotation, as each pair of teeth intermesh on the inlet side, the volume on that side is reduced by the volume of two tooth spaces, providing a suction effect. Oil following into the suction is then trapped on each side by tooth crest approaching the bore of the housing and carried round to the delivery side by the “pockets” between adjacent pairs of teeth. On the delivery side the oil is displaced from the delivery port under pressure. [38]

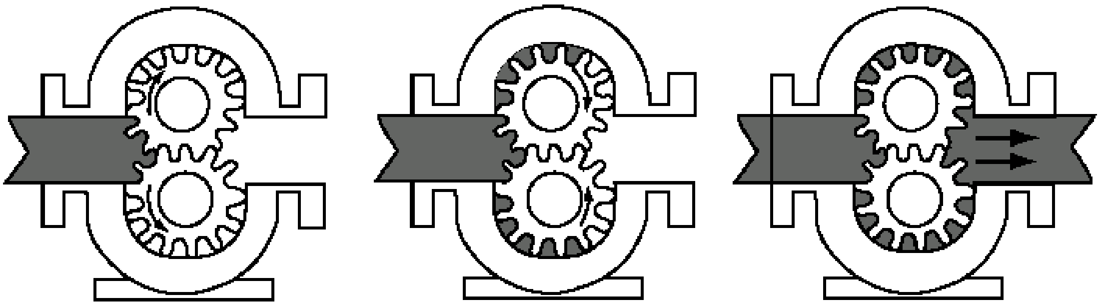


Figure (4.3) External Gear Pump

- **Internal Gear Pump**

The internal gear pump is based on a different geometry to that of an external gear pump; internal gear is located within an outer gear ring, the tooth form being chosen so each internal gear tooth is in contact with the inner surface of the outer ring as shown in figure (4.4). There are numerous design variations possible but all work on the principle of producing a sequence of expanding and contracting pockets into which liquid is drawn from the inlet side, transferred to the outside and forced out under pressure. This geometry has some specific advantages over the external gear pump, notably the lower localised fluid pressure generated and the lower shearing forces on the fluid. It can also show much lower operating noise levels than a comparable external gear pump and readily lends itself to multi-staging.

With simple gear forms, the necessary blocking action to prevent back flow from the outlets to the inlet side may be provided by the gears themselves by a crescent-shaped partition fitted to the case between the gears. Internal gear pumps are better suitable to handling higher viscosity and shear sensitive fluids, although they are equally suitable for handling low viscosity fluids and even volatile liquids with suitable shaft seals. They are not suitable for handling low viscosity fluids which are non lubricants or contaminated fluid. An advantage of the internal gear pump compared with an external gear pump is that it can produce larger displacements for the same overall size. [39]

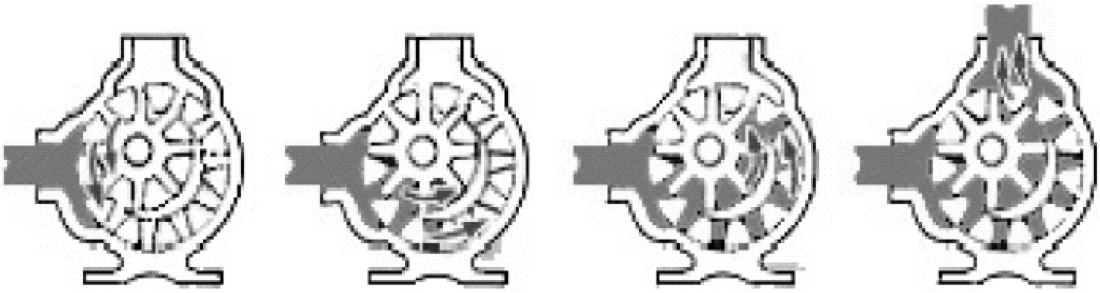


Figure (4.4) Internal Gear Pump

4.2.1.2.2. Lobe Pumps

Lobe pumps operate on a similar principle to internal gear pumps except that the opposite rotating intermeshing elements are of lobe rather than gear form and maintain a small but positive clearance at times. Both rotors are power driven and synchronised gear or a timing chain mounted externally, as shown in figure (4.5).

The considerable variety of rotor forms which can be employed also provides control over efficiency, suitability for handling a particular product and capacity. Lobe rotor pumps are also suitable for handling gases as well as liquids, and are thus also used as compressors and exhausters.

Plain rotor forms are generally suitable for handling a wide variety of fluid viscosities, two-lobe, three or four-lobe rotors generally being preferred for straightforward duties.

Whilst capacity is directly proportional to speed, it is rather more dependent on head than in the case of most other positive rotary pumps. For high head working recovery of capacity by increasing the face width of the rotors considerably increases the load on the pump components and so the normal practice is to decrease the face width for high head lobe rotor pumps and go to a larger rotor diameter, if only to reduce the overhung load on the bearings. [38]

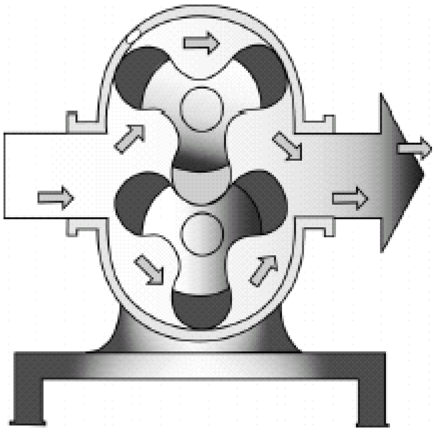


Figure (4.5) Lobe Pump

4.2.1.2.3. Vane Pumps

The vane pump itself consists of a circular rotor with radial slots mounted eccentrically in a substantially circular casing. Each rotor slot carries a rigid vane, free to slide a radial direction, as shown in figure (4.6) Rotation throws the vanes outwards so throws the tips always rub against the inner of the casing providing a seal. The actual shape of the casing and the eccentric location of the rotor inlet port communicate with the casing on the expanding volume side and an outlet port connects on the contracting valve side. Fluid is thus sucked in through the inlet port and squeezed out under pressure through the outlet port. In pumps for liquids the ports completely cover the whole inlet and outlet. Where air or gas is being pumped it may be desirable to reduce the port size to achieve a higher degree of compression on the delivery side.

There are various types of rigid rotor vane pumps where movable sealing elements in the form of rigid blades are moved radically inward and outward by cam surfaces to maintain a fluid seal between the open to inlet volume and the open to outlet volume during the operation of the pump. When the cam surface is internal to the body member and vanes are mounted in or on the rotor, the pump is called an internal vane pump or vane in rotor pump. When the cam surface is the external radial surface of the rotor and the vanes are mounted in the body or stator, the pump is known as an external vane pump or vane in body pump. [39]

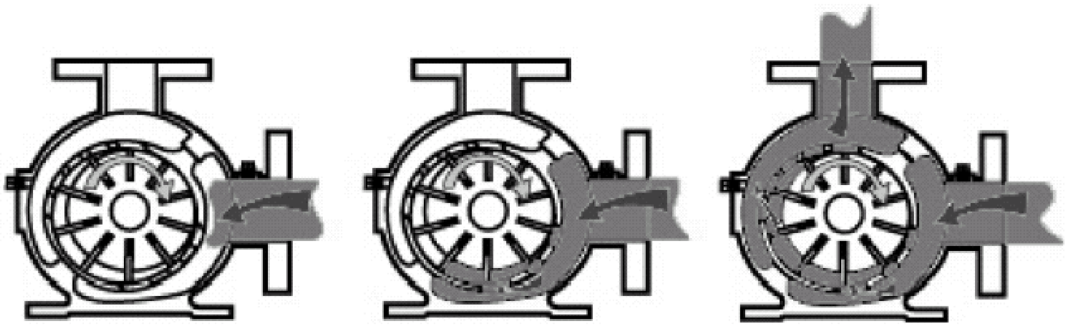
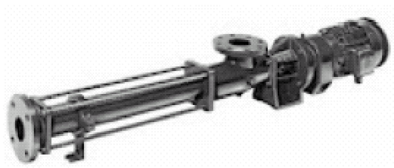


Figure (4.6) Vane Pump

4.2.1.2.4. Progressive Cavity Pumps

A progressive cavity pump is designed specifically to transfer abrasive and viscous fluids with a high solid, fibre and air content. As the rotor turns, it contacts the stator along a continuous sealing line, creating a series of sealed cavities that progress to the discharge end, as shown in figure (4.7), the cavity fills with liquid as it gradually opens and expands at the suction end of the rotor. The trapped liquid is transported to the discharge end and is then gradually discharged in an axial direction. Multistage pumps of up to four stages are available for reduced wear from abrasives.

Progressive cavity pumps are used in wastewater treatment plants for transferring all types of slurries and sludge. These pumps are self-priming but they cannot be operated dry. The flow is even and the shear is low. The pump is relatively easy to service. Progressive cavity pumps are relatively low cost, but stators and rotors may have to be replaced frequently, especially if grit is present in the fluid. [41]



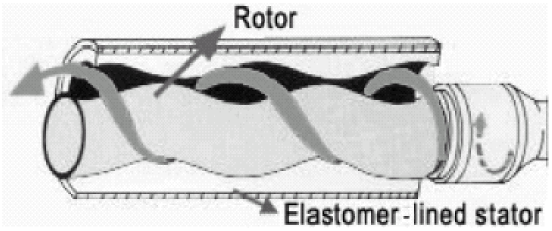


Figure (4.7) Progressive Cavity Pump

4.2.1.2.5. Screw Pumps

Screw pumps are a special type of rotary positive displacement pump in which the flow through the pumping elements is truly axial. The liquid is carried between screw threads on one or more rotors and is displaced axially as the screws rotate and mesh, as shown in figure (4.8). In all other rotary pumps the liquid is forced to travel circumferentially, thus giving the screw pump with its unique axial flow pattern and low internal velocities. The screw pump can handle liquids in range of viscosity from molasses to gasoline, as well as synthetic liquids. Because of the relatively low inertia of their rotating parts, screw pumps are capable of the operating at higher speed than other rotary or reciprocating pumps of comparable displacement. [3]

Screw pumps are classified into single or multiple rotor types.

- The single screw pump exists only in a limited number of configurations. The rotor thread is eccentric to the axis of rotation and meshes with internal threads of the stator; alternatively the stator is made to wobble along the pump center line.
- Multiple screw pumps are available in a variety of configurations and designs. All employ one driven rotor in mesh with one or more sealing rotors.

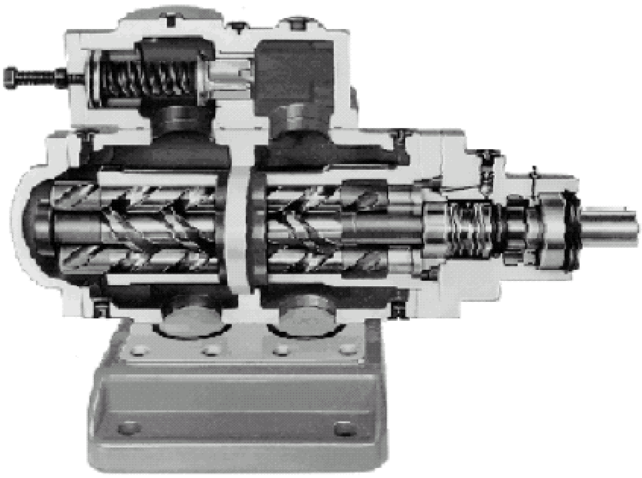


Figure (4.8) Three Spindle Screw Pump

4.2.2. Centrifugal Pumps

Centrifugal pumps are more common when large volumes must be pumped and high pressure differentials are not present. Rather than operating with a reciprocating motion, the centrifugal pump rotates. It consists of an impeller and a casing. The impeller is turned by the pump's driver through a shaft and throws the liquid into the pump casing, increasing the energy of the liquid by centrifugal force. This increase in energy causes the liquid to flow into the discharge line. Movement of the liquid out of the impeller reduces the pressure at the impeller inlet, allowing more fluid to flow into the impeller from the suction line.

There is a variety of types of centrifugal pumps, and capacities cover a wide range. Some pumps have one impeller and casing; others have several impellers and casing arranged in series. Different casing designs are also available, casings can be either one split or two split. Some centrifugal pumps are mounted along their centreline, inline pumps are used primarily in plant applications rather than in pipeline service.

The location of the suction, or inlet to the pump, also distinguishes two types of centrifugal pumps. The inlet is located axially, concentric with the impeller, in end suction pumps. In side suction pumps, the suction inlet is perpendicular or the axis of the impeller.

In addition to capacity, centrifugal pumps are classified according to their specific speed, which relates flow rate, head, and operating speed of the pump. Figure (4.9) shows the existing centrifugal pumps which are used at Ghani oil field.

Figure (4.9 A) 4 x 11 MSNM 5 Stages Centrifugal Pump (255 Kw)

Figure (4.9 B) 3 x 9 WMSN-H 6 Stages Centrifugal Pumps (150 Kw)

4.2.3. Oil Separators

The crude oil produced from the oil production wells flows to a station production manifold through the connected flow lines between the wellhead and the production manifold, as the crude oil contains a mixture of oil, water and gas. The crude oil is sent to three phases production separator where the separation take place by gravitation force and demulisifier chemical is added to the system to help the water separation, then the separated water is sent to the separate tank for further treatment and re-injected into the reservoir in order to maintain the servitor pressure. Thus the gas liberated from the top of the production separator part of this gas is dehydrated and utilized to be used at the station as fuel gas for turbine and diesel plant heaters, and the remaining gas produced is sent to flare as unwanted gas.

The oil produced from the production separator is heated up and goes to a desalater where the oil is washed by water with lower salinity to remove all partial of the salt contents, and all dissolved water and gas is removed from the oil. Purified oil is transferred to oil storage tanks ready to be for exported through the oil metering system and oil transmission pipeline to Ras Lanuf Terminal as final oil product.

4.2.4. Storage Tanks

Storage facilities for crude and natural gas products are an important element in all pipeline and tanker transportation systems. Storage allows flexibility in pipeline and refinery operations and minimizes unwanted fluctuations in pipeline throughput and product delivery.

Oil from individual wells is accumulated in tanks and then pumped into the crude oil gathering pipeline. If the crude is run manually, an operator will start the pump when the tank is full and will stop it when the tank is nearly empty.

Crude storage tanks are cylindrical and are operated at near atmospheric pressure. Small storage tanks are typically shop-fabricated and are delivered to the site where they are connected to pumps. Large crude storage tanks may be capable of storing up to several hundred thousand barrels each and must be built on the site. Large crude storage tanks often have a floating roof that moves up and down with the liquid level in the tank to minimize vapour losses. Smaller storage tanks have fixed roofs.

As discussed earlier in chapter 1. The Ghani oil field includes five stations, Ghani station is the main, Facha station, Tagrifet station, Zenad station and Eddib station. As shown in figure (4.10), the crude oil goes from Facha station, Tagrifet station, Eddib station, manifold *RRR*17 and Manifold *RRR*13 to Zenad station, then directly to the main station at Ghani field. Crude oil is transported from Ghani oil field to Joffra oil field then from Joffra oil field to Ras Lanuf Terminal.

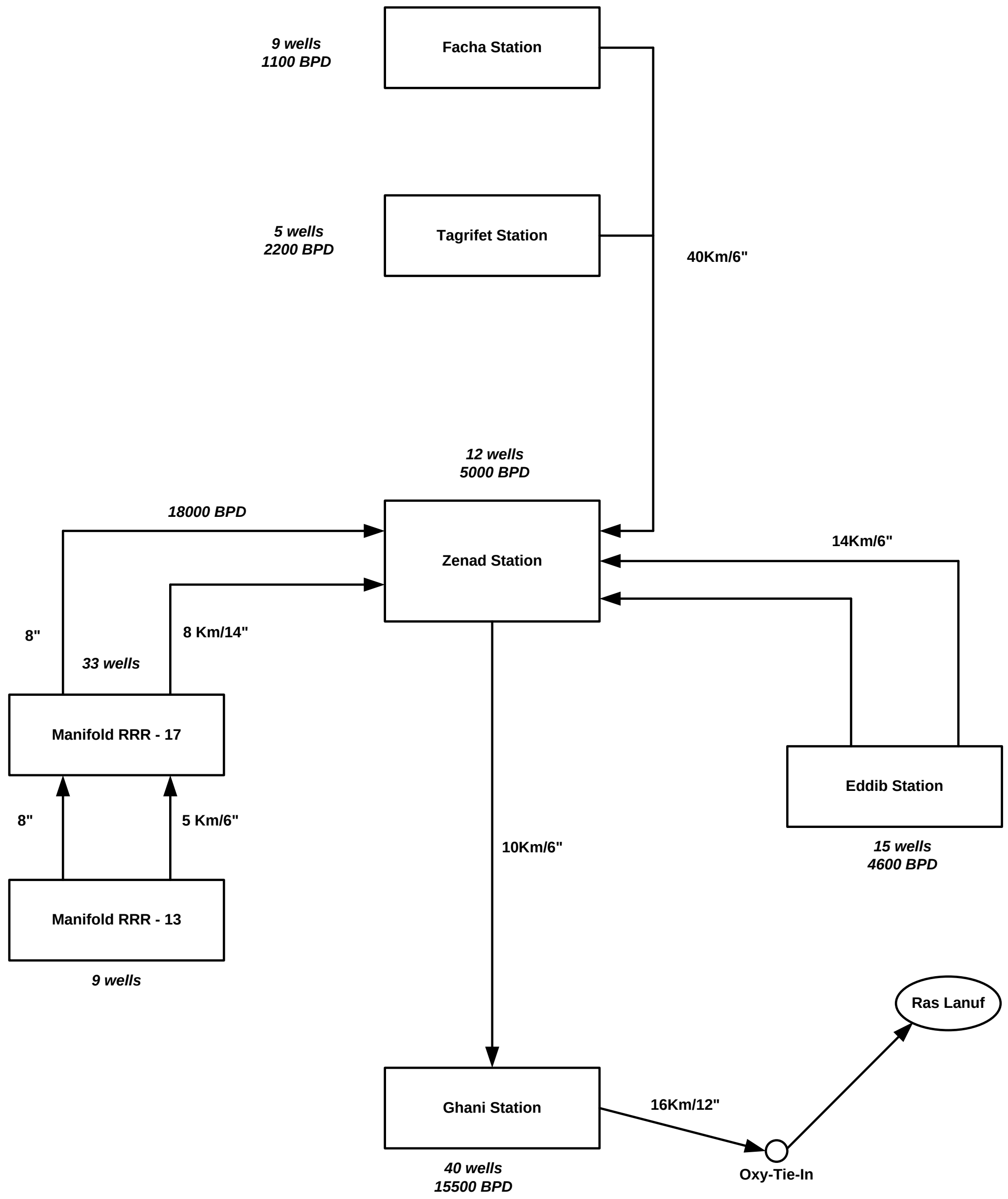


Figure (4.10) Productive Stations at Ghani Field

4.3. Pumps Performance

Pumping can be defined as the addition of energy to a move it from one point to another, it is not, as frequently thought, the addition of pressure. Because energy is capacity to do work, adding it to a fluid causes the fluid to do work, normally flowing through a pipe or rising to a higher level.

A centrifugal pump transforms mechanical energy from a rotating impeller into the kinetic and potential energy required. Although the centrifugal force developed depends on both the peripheral speed of the impeller and the density of the fluid, the amount of energy imparted per mass unit of fluid is independent of the fluid itself. Therefore, for a given machine operating at certain speed and handling a definite volume, the mechanical energy applied and transferred to the fluid is the same for any fluid, regardless the density. The pump head will therefore be expressed in meters. Barring viscosity effects, the head generated by a given pump at a certain speed and capacity will remain constant for all fluids. [45]

When pumps are selected and installed, good engineering dictates a careful study of conditions to ensure obtaining efficient equipment and economical size and arrangement of piping. If piping head losses are excessive, the pump must operate at reduced capacity, and efficiency and the power consumption will be high. These objectionable and costly conditions can be avoided by simple methods of analysis, which include combining pump and pipeline friction head characteristics.

Performance of a centrifugal pump can be readily understood from its characteristic curves, such as in figure (4.17), supplied by the manufacturer. The capacity-head curve shows the relation between cubic meters per hour (m^3/h) and total head in meter (m) against which a given capacity can be discharged. This curve definitely fixes the pump's operating range, and it must operate at some point along the curve.

The pump efficiency is 77 per cent; the efficiency curve is obtained by plotting per cent of pump efficiency against cubic meters per hour discharged, covered by the pumping range. The brake horsepower required to drive the pump when delivering the capacities indicated.

4.3.1. Effect of Temperature

Centrifugal pumps are employed at present for liquids at temperatures up to 850 ^{0}F. (454 ^{0}C) Shaft deflections are inversely proportional to the modulus of elasticity, and hence the critical speed will be lower at high temperatures.

Another effect of temperature on critical speeds is observed when the degree of stiffening of the shaft by the impeller hubs and sleeves changes as a result of difference in the heat expansion of the shaft and parts mounted on the shaft. [9]

4.4. Pumping Systems

The main items are pumps and their drivers, many pumping stations have several pumps. The number depends on the total horsepower required and the individual capacity ratings of each pump, it can also depend on the designer's preference. (Table 4.1 shows M.O.L. centrifugal pumps)

Pump stations typically include metering equipment for measuring throughput. Major stations, where custody of the fluid is transferred from one point to another, contain a meter prove to calibrate the metering equipment. Originating stations may be also have storage tanks to smooth out variations in flow to the station so the pump will operate continuously at near normal capacity, even though small changes in the supply of crude or products to the station occur. In addition to the M. O. L. centrifugal pumps in Ghani station with its storage tanks, there are four booster pumps included to move liquid from storage tanks to the suction line of the M. O. L. centrifugal pumps.

type	Horizontal, Centrifugal Pumps
Size & Type	$4\times$ 11 MSNM
Manufacturer	United Centrifugal Pumps Limited
Liquid	Crude Oil
Pumping Temperature	130 ^{0}F (54.4 ^{0}C)
Capacity	876 G.P.M
Suction Pressure	60 Psig
Discharge Pressure	500 Psig
Differential Head	647 m
NPSH Available	26 m
NPSH Required (Water) 3% Head Drop	8.5 m
Suction Specific Speed	7197
Specific Gravity	60 ^{0}F (15.5 ^{0}C) 0.832
Efficiency	0.77
Break Horsepower	342 HP
Speed	2960 R.P.M
Nozzle Size	Suction 6”- Discharge 4”
No. of Stage	5 Stages
Shaft Seal	Mechanical Seal
Driver Type	Electrical Motor
Type	KR 5029 B-DAO2
Manufacturer	Schorch GMBH
Rated Speed	2960 R.P.M
Rated Output	255 Kw
Voltage	6600 V - 3 PH – 50 HZ

Table (4.1) M.O.L. Centrifugal Pumps Data Sheet [40]

4.4.1. Engineering of System Requirements

The first decision which should be made is to determine the requirements and conditions under which the equipment will operate is the consumption of energy.

Pumps are not the biggest consumers of energy, but they are not the smallest either. The main three points to remember are that slower flows mean lower pump horsepower; slower flows also mean improved pump station efficiency via greater differentials, since pump horsepower is directly proportional to the cube of flow rate. Further the cost of the pumping system can be reduced by decreasing the minor and major losses in the piping system and to use other pumps.

4.4.1.1. Hydraulic Design and Calculations

The hydraulic calculations for the pumping station have been discussed in Chapter 3.

In general, hydraulic design is the process of evaluating the physical characteristics of the product or commodities to be transported, the quantities to be transported, the pipeline route and topography, the range of pressures, temperatures, and the environmental conditions along the route.

The elevation of the hydraulic gradient into the station should be represented above the ground elevation by an allowance to supply station losses to the pumps, and the required net positive suction head (*NPSH*) at the pump.

In consequence:

Total system losses = Total suction losses + Total discharge losses $\pm$ Elevation head

$$h_{FS} = h_l + h_f \qquad (4.1)$$

Where:

h_{FS} = Total suction losses, (m)

h_l = Losses due to friction line, (m)

h_f = Losses due to fittings, (m)

$$H_f = f \frac{L \times V^2}{D \times 2 \times g} \tag{4.2}$$

$$V = \frac{Q}{A} = \frac{Q \times 4}{\pi \times D^2} \tag{4.3}$$

$$f = \frac{0.25}{\left[\log\left(\frac{K}{3.7D} + \frac{5.74}{\mathrm{Re}^{0.9}}\right)\right]^2} \tag{4.4}$$

Where:

K = Roughness value, (4.52×10^{-5} m), for commercial steel, from table (3.1)

D_S = Inner diameter (m), suction line

D_D = Inner diameter (m), discharge line

R_e = Reynolds number, $\frac{VD}{v}$

v = Kinematics viscosity for the discharge line (8.1×10^{-6} m^2/sec)

L_S = Length of the suction line (205m)

L_D = Length of the discharge 81.25 miles, (130×10^3 m)

4.4.1.1.1. Losses due to Fittings

Invariably a system containing piping will have connections which change the size and/or direction of the pipe. These fitting losses together with friction losses are called minor losses to the system head. Fitting losses are generally the result of changes in velocity and/or direction. A decrease in velocity results in more loss in head than an increase velocity as the former causes energy-dissipating eddies. Experimental results have indicated that minor losses vary approximately as the square of the velocity at entry to the fittings.

The resistance to flow through valves and fittings may be estimated by any of the following methods. Losses through valves and fittings in terms of the average velocity head in the pipe of the corresponding diameter and a resistance coefficient, the fractional resistance in (m) is found from the equation: [3]

$$h_f = k \times \frac{V^2}{2 \times g} \qquad (4.5)$$

Where:

h_f = Losses due to fittings, (m)

k = Resistance coefficient, tables (4.2), (4.3), (4.4), (4.5) and (4.6)

V = Average pipe velocity (m/sec)

g = Acceleration of gravity (9.81m/sec^2)

Losses due to valves and fittings in the discharge line will be a very small percentage of the total head and is often considered negligible in energy calculations. Figure (4.11) shows schematically the suction line of a pumping station.

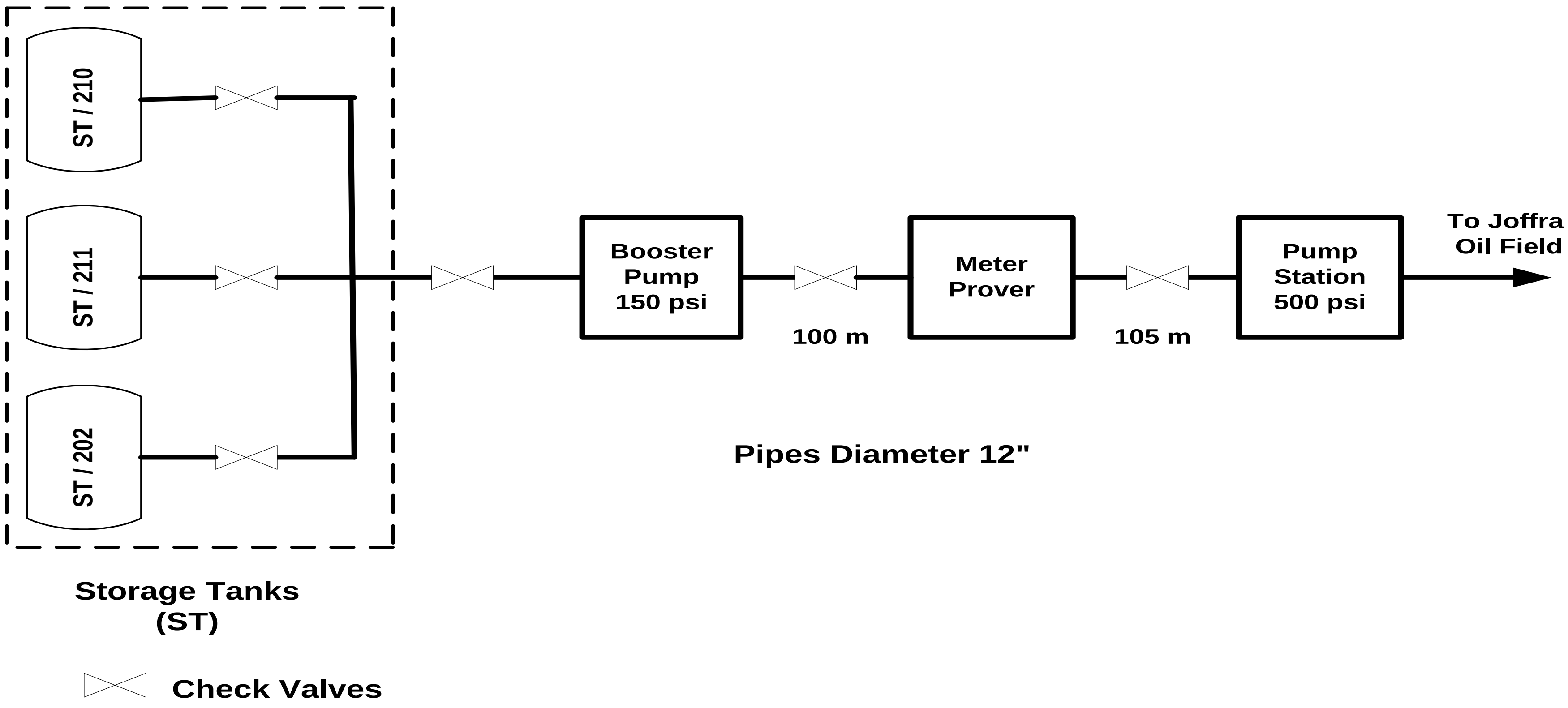


Figure (4.11) The Suction Line of the pumping Station

Fitting		L/D	Nominal pipe size ½	¾	1	1¼	1½	2	2½–3	4	6	8–10	12–16	18–24
			K value											
Butterfly Valve								0.86	0.81	0.77	0.68	0.63	0.35	0.30
Plug Valve straightway		18	0.49	0.45	0.41	0.40	0.38	0.34	0.32	0.31	0.27	0.25	0.23	0.22
Plug Valve 3-way thru-flo		30	0.81	0.75	0.69	0.66	0.63	0.57	0.54	0.51	0.45	0.42	0.39	0.36
Plug Valve branch-flo		90	2.43	2.25	2.07	1.98	1.89	1.71	1.62	1.53	1.35	1.26	1.17	1.08
Standard elbow	90°	30	0.81	0.75	0.69	0.66	0.63	0.57	0.54	0.51	0.45	0.42	0.39	0.36
	45°	16	0.43	0.40	0.37	0.35	0.34	0.30	0.29	0.27	0.24	0.22	0.21	0.19
	long radius 90°	16	0.43	0.40	0.37	0.35	0.34	0.30	0.29	0.27	0.24	0.22	0.21	0.19

Resistance coefficient K (use in formula $h_f = K\frac{V^2}{2g}$)

Table (4.2) Friction Losses in Fittings

Fitting	Type of bend	L/D	Nominal pipe size ½	¾	1	1¼	1½	2	2½–3	4	6	8–10	12–16	18–24
			K value											
Close Return Bend		50	1.35	1.25	1.15	1.10	1.05	0.95	0.90	0.85	0.75	0.70	0.65	0.60
Standard Tee	thru flo	20	0.54	0.50	0.46	0.44	0.42	0.38	0.36	0.34	0.30	0.28	0.26	0.24
	thru branch	60	1.62	1.50	1.38	1.32	1.26	1.14	1.08	1.02	0.90	0.84	0.78	0.72
90° Bends, Pipe bends, flanged elbows, butt welded elbows	r/d = 1	20	0.54	0.50	0.46	0.44	0.42	0.38	0.36	0.34	0.30	0.28	0.26	0.24
	r/d = 2	12	0.32	0.30	0.28	0.26	0.25	0.23	0.22	0.20	0.18	0.17	0.16	0.14
	r/d = 3	12	0.32	0.30	0.28	0.26	0.25	0.23	0.22	0.20	0.18	0.17	0.16	0.14
	r/d = 4	14	0.38	0.35	0.32	0.31	0.29	0.27	0.25	0.24	0.21	0.20	0.18	0.17
	r/d = 6	17	0.46	0.43	0.39	0.37	0.36	0.32	0.31	0.29	0.26	0.24	0.22	0.20
	r/d = 8	24	0.65	0.60	0.55	0.53	0.50	0.46	0.43	0.41	0.36	0.34	0.31	0.29
	r/d = 10	30	0.81	0.75	0.69	0.66	0.63	0.57	0.54	0.51	0.45	0.42	0.39	0.36
	r/d = 12	34	0.92	0.85	0.78	0.75	0.71	0.65	0.61	0.58	0.51	0.48	0.44	0.41
	r/d = 14	38	1.03	0.95	0.87	0.84	0.80	0.72	0.68	0.65	0.57	0.53	0.49	0.46
	r/d = 16	42	1.13	1.05	0.97	0.92	0.88	0.80	0.76	0.71	0.63	0.59	0.55	0.50
	r/d = 18	46	1.24	1.15	1.06	1.01	0.97	0.87	0.83	0.78	0.69	0.64	0.60	0.55
	r/d = 20	50	1.35	1.25	1.15	1.10	1.05	0.95	0.90	0.85	0.75	0.70	0.65	0.60
Mitre Bends	α = 0°	2	0.05	0.05	0.05	0.04	0.04	0.04	0.04	0.03	0.03	0.03	0.03	0.02
	α = 15°	4	0.11	0.10	0.09	0.09	0.08	0.08	0.07	0.07	0.06	0.06	0.05	
	α = 30°	8	0.22	0.20	0.18	0.18	0.17	0.15	0.14	0.14	0.12	0.11	0.10	0.10
	α = 45°	15	0.41	0.38	0.35	0.33	0.32	0.29	0.27	0.26	0.23	0.21	0.20	0.18
	α = 60°	25	0.68	0.63	0.58	0.55	0.53	0.48	0.45	0.43	0.38	0.35	0.33	0.30
	α = 75°	40	1.09	1.00	0.92	0.88	0.84	0.76	0.72	0.68	0.60	0.56	0.52	0.48
	α = 90°	60	1.62	1.50	1.38	1.32	1.26	1.14	1.08	1.02	0.90	0.84	0.78	0.72

Resistance coefficient K (use in formula $h_f = K\frac{V^2}{2g}$)

Calculated from data in Crane Co. Technical Paper No. 410.

Tables (4.2) & (4.3) From Cameron Hydraulic Data [19]

Fitting stop-check valves	L/D	Minimum velocity for full disc lift: general ft/sec†	water ft/sec	Nominal pipe size ½	¾	1	1¼	1½	2	2½–3	4	6	8–10	12–16	18–24
				K value*											
	400	$55\sqrt{\bar{V}}$	6.96	10.8	10	9.2	8.8	8.4	7.5	7.2	6.8	6.0	5.6	5.2	4.8
	200	$75\sqrt{\bar{V}}$	9.49	5.4	5	4.6	4.4	4.2	3.8	3.6	3.4	3.0	2.8	2.6	2.4
	350	$60\sqrt{\bar{V}}$	7.59	9.5	8.8	8.1	7.7	7.4	6.7	6.3	6.0	5.3	4.9	4.6	4.2
	300	$60\sqrt{\bar{V}}$	7.59	8.1	7.5	6.9	6.6	6.3	5.7	5.4	5.1	4.5	4.2	3.9	3.6
	55	$140\sqrt{\bar{V}}$	17.7	1.5	1.4	1.3	1.2	1.2	1.1	1.0	.94	.83	.77	.72	.66

Resistance coefficient K (use in formula $h_f = K\frac{V^2}{2g}$)

Table (4.4) Friction Losses in Pipe Fittings

Fitting	L/D	Minimum velocity for full disc lift: general ft/sec†	water ft/sec	Nominal pipe size ½	¾	1	1¼	1½	2	2½–3	4	6	8–10	12–16	18–24
				K value*											
Swing check valve 	100	$35\sqrt{\bar{V}}$	4.43	2.7	2.5	2.3	2.2	2.1	1.9	1.8	1.7	1.5	1.4	1.3	1.2
	50	$48\sqrt{\bar{V}}$	6.08	1.4	1.3	1.2	1.1	1.1	1.0	0.9	0.9	.75	.70	.65	.6
Lift check valve 	600	$40\sqrt{\bar{V}}$	5.06	16.2	15	13.8	13.2	12.6	11.4	10.8	10.2	9.0	8.4	7.8	7.2
	55	$140\sqrt{\bar{V}}$	17.7	1.5	1.4	1.3	1.2	1.2	1.1	1.0	.94	.83	.77	.72	.66
Tilting disc check valve 	5°	$80\sqrt{\bar{V}}$	10.13						.76	.72	.68	.60	.56	.39	.24
	15°	$30\sqrt{\bar{V}}$	3.80						2.3	2.2	2.0	1.8	1.7	1.2	.72
Foot valve with strainer poppet disc 	420	$15\sqrt{\bar{V}}$	1.90	11.3	10.5	9.7	9.3	8.8	8.0	7.6	7.1	6.3	5.9	5.5	5.0
Foot valve with strainer hinged disc 	75	$35\sqrt{\bar{V}}$	4.43	2.0	1.9	1.7	1.7	1.7	1.4	1.4	1.3	1.1	1.1	1.0	.90

Resistance coefficient K (use in formula $h_f = K\frac{V^2}{2g}$)

Tables (4.4) & (4.5) Friction Losses in Pipe Fittings [19]

Fitting	L/D	Nominal pipe size ½	¾	1	1¼	1½	2	2½–3	4	6	8–10	12–16	18–24
		K value											
Gate Valves	8	0.22	0.20	0.18	0.18	0.15	0.15	0.14	0.14	0.12	0.11	0.10	0.10
Globe Valves	340	9.2	8.5	7.8	7.5	7.1	6.5	6.1	5.8	5.1	4.8	4.4	4.1
Angle Valves	55	1.48	1.38	1.27	1.21	1.16	1.05	0.99	0.94	0.83	0.77	0.72	0.66
Angle Valves	150	4.05	3.75	3.45	3.30	3.15	2.85	2.70	2.55	2.25	2.10	1.95	1.80
Ball Valves	3	0.08	0.08	0.07	0.07	0.06	0.06	0.05	0.05	0.05	0.04	0.04	0.04

Resistance coefficient K (use in formula $h_f = K \frac{V^2}{2g}$)

Table (4.6) Friction Losses in Pipe Fittings [19]

Head losses (h_f) through valves, fitting, sudden contractions and enlargements, entrance and exit loss can be expressed in term of the velocity head ($V/2g$) by using the applicable resistance coefficient (k) in the equation (4.5).

Select applicable (k) from tables (4.2) to (4.6), and select average velocity (V) in the pipe of diameter required to accommodate fitting, the method of expressing head losses (h_f) through valves and fittings etc. is in terms of the equivalent length of straight pipe that will product the same losses.

The tables (4.2) to (4.6) list (k) values between pipes sizes are close, it is reasonable to interpolate between sizes if they do not correspond to diameters.

4.4.1.1.2. Net Positive Suction Head

One of the most important of several criteria in the design of a liquid pumping system is the net positive suction head (*NPSH*) required. Each pump has its own requirement for *NPSH*, usually expressed in (metres) of head. The *NPSH* required is normally shown on the manufacture's rating curve for each pump.

The net positive suction head available to a pump in a specific application must be equal to or greater than the required *NPSH* specified by the manufacturer. If enough suction head is not available at the desired flow rate, vapour lock, cavitation, and pump damage can result.

Basically, the net positive suction head is the difference in pressure between the tank and pipe from which liquid flows to the pump suction and the centre of the pump suction. *NPSH* is determined from the pressure in the tank or pipe, the atmospheric pressure, the vapour pressure of the liquid, the specific gravity of the liquid, the friction losses in the piping, valve losses, and the difference in elevation between the fluid in the tank and the pumping suction.[18]

Accordingly, the *NPSH* available is given by:

$$NPSH_A = h_a - h_{vap} - h_{FS} \pm h_{st} \qquad (4.6)$$

Where:

h_a = Absolute pressure, (psi)

h_{vap} = Vapour pressure, (psi)

h_{FS} = Line losses, (m)

h_{st} = Static elevation of the liquid above the entrance eye of the pump, (m)

Where:

Pressure in feet = $\dfrac{psi \times 2.31}{Sg}$

Pressure in (m) = $\dfrac{KPa \times 9.807}{Sg}$

Total system losses = Total suction losses + Total discharge losses $\pm$ Elevation head

Tables (4.7), (4.8) and (4.9) show how the change in capacity affects some factors such as velocity, friction factor and Reynolds number. They also illustrate the losses in the suction line, discharge line and the total system losses for four different flow rates during one year at Ghani oil field.

From table (4.7) the total system loss at flow rate 45,000 BPD is equal to the total suction losses + total discharge losses $\pm$ elevation head between the pumping station (Ghani station) and Oxy-Tie-In, the elevation at Ghani oil field is (260 m) and the elevation at Oxy-Tie-In is (130 m), so the elevation head is (-130 m). The total system loss is equal to 4.25 m (2.25 + 132 -130 = 4.25 m).

Also the velocity, friction factor, and Reynolds number at the same flow rate are equal due to the same pipe diameter 12 inch (0.305m). See figure (4.11).

Capacity		Velocity		Friction Factor		Reynolds No.	
BPD	m^3/h	suction m/sec	discharge m/sec	suction	discharge	suction	discharge
45,000	298	1.5	1.5	0.0208	0.0208	56481.5	56481.5
40,000	265	1.4	1.4	0.0211	0.0211	52716	52716
35,000	232	1.323	1.323	0.0214	0.0214	49817	49817
30,000	199	1.233	1.233	0.0222	0.0222	46428	46428

Table (4.7) Change of Capacity and its Affect on the Velocity, Friction Factor and Reynolds Number, from Ghani to Oxy-Tie-In (16 Km)

Capacity		Suction line			Discharge losses head	Total system losses
BPD	m^3/hr	line losses	Fitting losses	Total losses		
		m	m	m	m	m
45,000	298	1.6	0.65	2.25	132	4.25
40,000	265	1.44	0.56	2	111	-17
35,000	232	1.3	0.5	1.8	100	-28.2
30,000	199	1.16	0.44	1.6	90	-38.4

Table (4.8) Change of Capacity and its Affect on the Losses of Pumping Station, from Ghani to Oxy-Tie-In (16 Km)

Capacity		Velocity	Reynolds No.	Friction factor	Total system losses
BPD	m^3/hr	m/sec			m
45,000	298	1.5	56481.5	0.0208	4.25
40,000	265	1.4	52716	0.0211	-17
35,000	232	1.323	49817	0.0214	-28.2
30,000	199	1.233	46428	0.0222	-38.4

Table (4.9) Change of Capacity and its Affect on Total System Losses of Pumping Station, from Ghani to Oxy-Tie-In (16 Km)

Capacity		Velocity	Friction Factor	Reynolds No.
BPD	m^3/h	discharge m/sec	discharge	discharge
62,900	417	0.680	0.0218	42647
56,500	374	0.640	0.0221	40138
50,000	331	0.606	0.0224	38006
43,500	288	0.570	0.0227	35748

Table (4.10) Change of Capacity and its Affect on the Velocity, Friction Factor and Reynolds Number, from Zuetina Zella and Ghani to Joffra.

Capacity		Discharge losses head	Total system losses
BPD	m^3/hr	m	m
62,900	417	114	284
56,500	374	103	273
50,000	331	93	263
43,500	288	84	254

Table (4.11) Change of Capacity and its Affect on the Losses of Pumping Station, from Zuetina Zella and Ghani to Joffra

Capacity		Velocity	Reynolds No.	Friction Factor	Total system losses
BPD	m^3/hr	m/sec			m
62,900	417	0.680	42647	0.0218	284
56,500	374	0.640	40138	0.0221	273
50,000	331	0.606	38006	0.0224	263
43,500	288	0.570	35748	0.0227	254

Table (4.12) Change of Capacity and its Affect on Total System Losses of Pumping Station, from Zuetina Zella and Ghani to Joffra

Tables (4.10), (4.11) and (4.12) show how the change in capacity affects some factors such as velocity, friction factor and Reynolds number. They also illustrate the losses in the discharge line and the total system losses for four different flow rates during one year at Ghani oil field and Zuetina Zella field.

From table (4.11) the total system loss at flow rate 62,900 BPD (417 m^3/h) between Zuetina Zella field 17,900 BPD (119 m^3/h) and Ghani oil field 45,000 BPD (298 m^3/h) is equal to the total suction losses + total discharge losses $\pm$ elevation head between the Oxy-Tie-In and Joffra oil field. The elevation at Oxy-Tie- In is (130 m) and the elevation at Joffra oil field is (300 m), so the elevation head is (170 m). The total system loss is equal to 284 m (0 + 114 + 170 = 284 m).

4.4.1.1.3. System Head Curve

The system head curve is a plot of total system resistance, variable plus fixed, for various flow rates, and has many uses in centrifugal pump application. It is preferable to express the system head in meters rather than Ib/in^2 since centrifugal pumps are rated in meters as previously explained.

When the system head curve is required for several flows, or when the pump flow is to be determined, a system head curve is constructed using the following procedure:

- Defined the pumping system and its length.
- Calculate (or measure) the fixed system head, which is the net change in total energy from the beginning to the end of the system, due to elevation and/or pressure-head difference. An increase in head in the direction of the flow is a positive quantity.
- Calculate the variable system total head loss through all pipes, valves, fittings and equipment in the system for several rates of flow. As shown in finger (4.12).

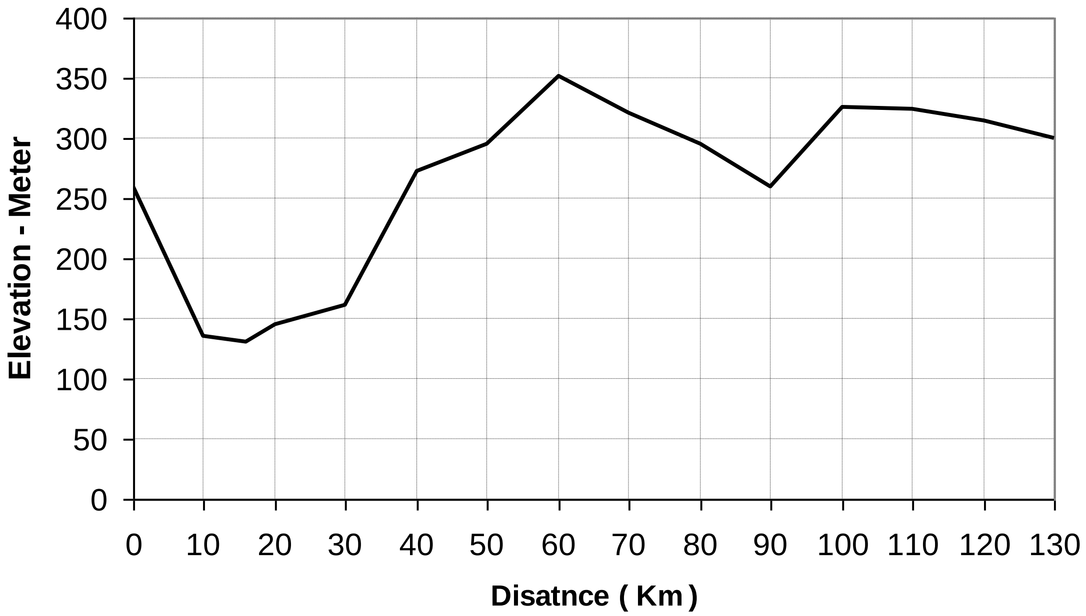
In this analysis, the pumping system is defined as that starting from the Ghani oil field to the Joffra oil field.

The variable system head is due to losses from pipe friction and valves and fittings.

The fixed head and variable heads for several flow rates are added together resulting in a total system head curve vs. flow. [3]

In this case the term loss due to fittings (minor loss) is negligible because its value is indeed very small.

Pipeline Route Profile



Ghani Joffra

Figure (4.12) Pipeline Route Profile – Ghani to Joffra fields

The calculation of the system head curve for the first section (Ghani field) of pipeline to Joffra field (Section 2) is given by the following equation:

$$Hf = f\,\frac{L \times V^2}{2 \times D \times g} \qquad \text{Equation (3.1)}$$

Therefore, the equivalent length of section 2 (Joffra field) based on diameters is given by: [20]

Equivalent Length $L_{e,}$ $= L_2 \left(\dfrac{D_1}{D_2}\right)^5$ (4.7)

$D_{1\,\text{Ghani}}$ = 12 inch (0.305m)

$D_{2\,\text{Joffra}}$ = 20 inch (0.508m)

$L_{1\,\text{Ghani}}$ = 16 Km

$L_{2\,\text{Oxy-Tie-In to Joffra}}$ = 113 Km

Equivalent Length

$$L_e = L_2\left(\frac{D_1}{D_2}\right)^5 = 113{,}000\left(\frac{0.305}{0.508}\right)^5 = 8{,}816 \text{ m}$$

Ie: $L_e = L_1 + L_2 = 16{,}000 + 8{,}816 = 24{,}816$ m

Where:

$f = 0.0208$

$D = 0.295$ m

$L = 24{,}816$ m

$$H_f = 0.0208 \times \frac{24{,}816 \times V^2}{2 \times 0.295 \times 9.81} = 89 \times V^2$$

But from the discharge equation

$$V = \frac{Q}{A} \qquad and \qquad V^2 = \frac{Q^2}{A^2}$$

$$A = \frac{\pi \times D^2}{4} = \frac{\pi(0.295)^2}{4} = 0.068 \quad m^2$$

Using these values we can calculate:

$$H_f = 89 \times V^2 = 89 \times \frac{Q^2}{(0.068)^2} = 19{,}247 \quad Q^2 \tag{4.8}$$

$$H_f = 19{,}247 \quad Q^2 + 40 \quad (Elevation \quad head) \tag{4.9}$$

From the above calculations, the head at different capacities can be obtained as tabulated in table (4.13) and the system head curve can be drawn as figure (4.13) below.

Discharge	m^3/hr	0	80	149	160	174	199	224	240	320	400	480
Head, H_f	m	40	49.5	73	78	85	99	115	125.5	192	278	382

Table (4.13) Value of Differential Head at Different Flow Rates

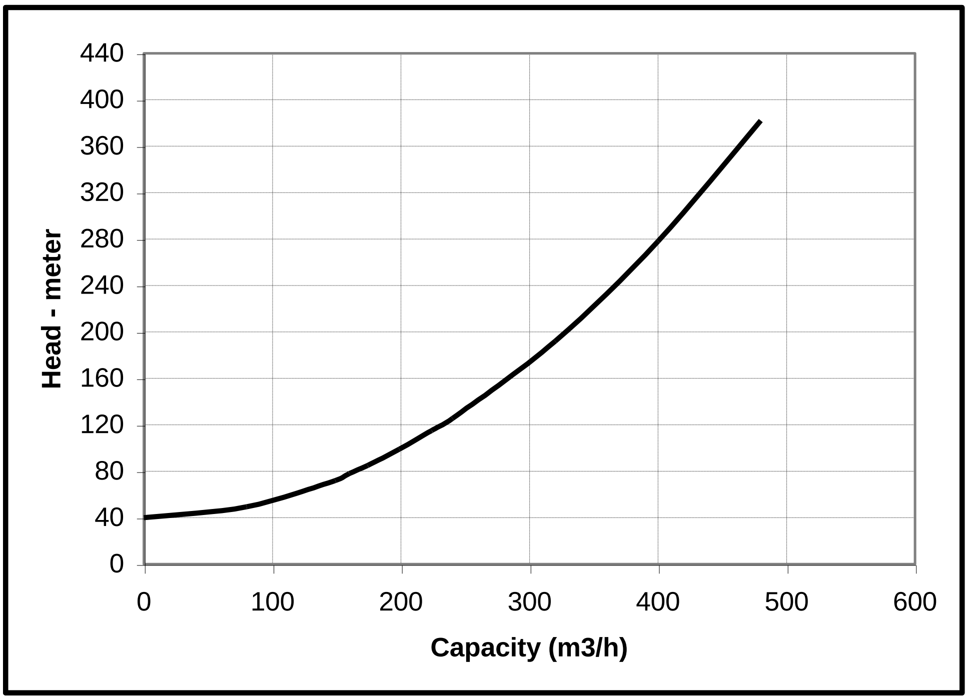


Figure (4.13) System Head Curve from Ghani oil field to Joffra oil field

4.4.1.1.4. Power Requirements and Costs for the System

The pumping station will be at the beginning of the pipeline system. The total pumping horsepower requirements for the complete system, known as brake horsepower (*BHP)*, can be determined using the following equation, where H_T, refers to the total system head requirements.

$$BHP = \text{flow, BPOD} \times \frac{H_T(feet) \times Sg}{136{,}000 \times Pump \quad efficiency} \tag{4.10}$$

$$BHP\ (\text{Kw}) = BHP \times 0.746 \tag{4.11}$$

The efficiency of the pump typically ranges from (75% to 85%) for centrifugal pumps. The determination of the brake horsepower for another station can be found by using the head or pressure required for the downstream section between one station and the next. The motor horsepower will exceed the break horsepower and can be calculated using the following equation:

$$\text{Power per pumping station} = \frac{BHP \quad (Kw)}{Motor \quad Efficiency} \tag{4.12}$$

The motor efficiency is assumed to be 83% and usually ranges between (80% to 90%).

Annual power cost per year = number of operating hours per year (5110) $\times$ Power per pumping station $\times$ cost per kwhr

Accordingly the annual power costs at different capacities of pumping station are:

Capacity		BHP	BHP(Kw)	Power per pumping Station	Annual power cost per year, at $ 0.07 / Kwhr
BPD	m^3/hr				
45,000	298	335	225	271	193873.4
40,000	265	297	222	267.5	191369.5
35,000	232	260	194	234	167403.6
30,000	199	223	166	200	143080

Table (4.14) Estimated Annual Power Cost for all the System

4.5. Pump Stations

The pumping capacity required at a given station must be based on a thorough overview of the proposed system. Such factors as the projected average and maximum day demands, safe demands of the available supply, and the function of the pump station within the total system operation must be considered.
Limitations on pumping capacity may be imposed by the demand of the oil supply. It is sometimes desirable to select the size of pumps to deliver only the safe demand of the source.

4.6. Selection of the Pump and its Driver

4.6.1. Pump Selection

Selecting an appropriate pump application is extremely important for best overall efficiency and reliability. Centrifugal pumps represent the largest category of pumps since they may be applied for clear low-viscosity liquid such as oil, water. The major influences on centrifugal pump efficiency are specific speed (*Ns*), pump size, $NPSH_A$ or ($NPSH_R$) and the type of pump selected to meet the service conditions. The efficiency of pumps typically ranges between 75% - 85%. To achieve the minor cost losses, the relative quantity of discharge and power is needed.

4.6.1.1. Impeller Specific Speed

Specific speed is a dimensionless factor that defines the impeller geometry and best attainable efficiency for the head, flow rate and speed.
The general shape of the impeller determines the field of application of a centrifugal pump. The discharge, head speed is chosen to give optimum performance for maximum efficiency.
The specific speed defines a suitable operating range. Figure (4.14) show typical impeller shapes relative to their specific speed.

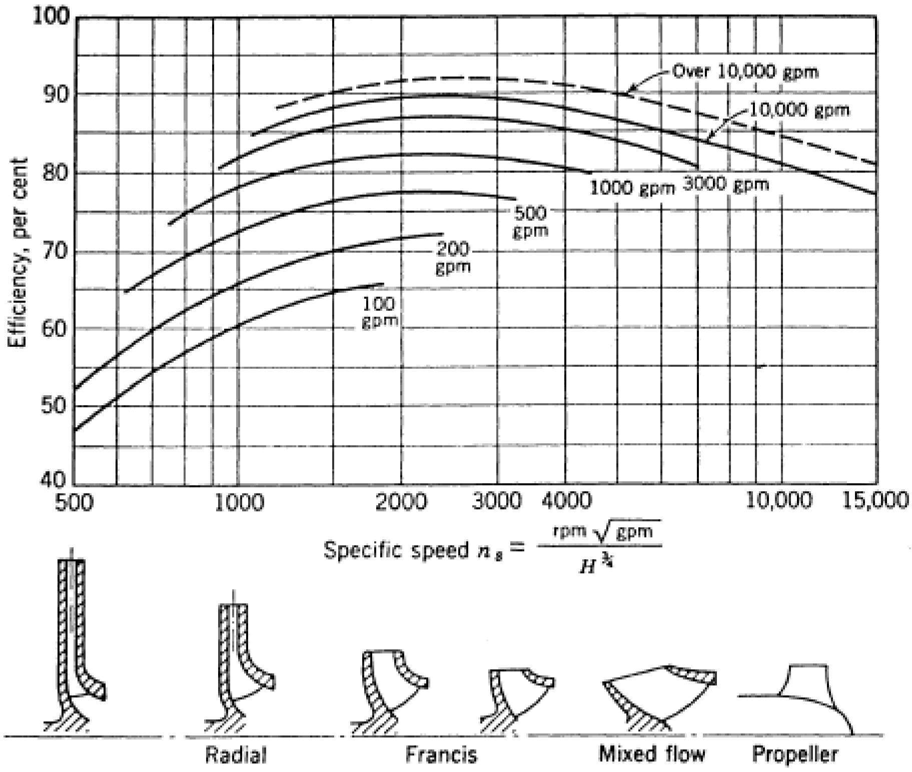


Figure (4.14) Typical Impeller Shapes Relative to Their Specific Speed [23]

Specific speed (N_s) is defined by the following equation.

$$Ns = \frac{N \times \sqrt{Q}}{H^{0.75}} \tag{4.13}$$

Where:

N_s = Specific speed, (rpm)

N = Rotational speed of the shaft, (rpm)

Q = Discharge, (m^3/h)

H = Total head, (m)

Below 4000 - radial flow,

Between 4000 and 9000 – mixed flow,

Above 9000 – axial flow,

4.6.1.2. Suction Specific Speed

More pumping installations fail because of poor suction conditions than from any other single use. A centrifugal pump has no capability to "suck" from a lower level, such as a well, unless it is initially primed and all the air is removed. A centrifugal pump is a kinetic energy machine designed to accelerate a volume of water from a low to a high velocity, and to convert this velocity into developed head at the pump discharge.
The suction specific speed is given by:

$$S = \frac{N \times \sqrt{Q}}{(NPSH_R)^{0.75}} \tag{4.14}$$

From experience reasonable values of suction specific speed (S) for hydrocarbon handling have been found to be in the range of 7000 to 15000.[19]

4.6.2. Driver Selection

The most common pump drives are AC electric motors. The key considerations in selecting electric motors for a pipeline prime mover application are often the cost and availability of electric power. In general the capital and annual costs for electric motors are lower when compared with a gas turbine, diesel and internal combustion engines. Due to availability of the electric power at the pump stations location, the electric motor would generally be the optimum selection.[18]

4.7. Pumps Connection

The system is required to handle different capacities during the life of the project, and also according to the minimum number of pump requirements in the station, it is advantageous to install several pumps in the pumping station. Pumps installed together may operate in series or in parallel. It is advantageous to install pumps of identical size. In this case their matching is best from a hydraulic standpoint. Also, from a practical standpoint, the cost of stocking spares will be less and interchangeability of the various components will facilitate repair and maintenance. An additional pump of similar size may also be added to the system. This is put on line when any one of the other pumps needs repair or maintenance.[1]

4.7.1. Pumps in Parallel and Series

Pumps can be connected in different ways to provide a range of operating conditions and capabilities. In a parallel arrangement as shown in figure (4.15), more than one pump takes suction from a single source. A suction manifold, consisting of a pipe from which individual suction lines branch off to the inlet of each pump, then each pump discharges separately into a discharge manifold connected to the pipeline. When connected in parallel, each pump operates at approximately the same suction and discharge pressure, and the total flow volume is sum of the output of the individual pumps.

Pumps can also be connected in series as shown in figure (4.16). In this case, one pump takes suction from the fluid source, then discharges into the pipeline. The suction pressure for the second pump is equal to the discharge pressure of the first pump minus losses in the connecting piping. The total flow volume is handled by each pump, but the total differential head is the sum of the differential heads of the individual pumps.[18]

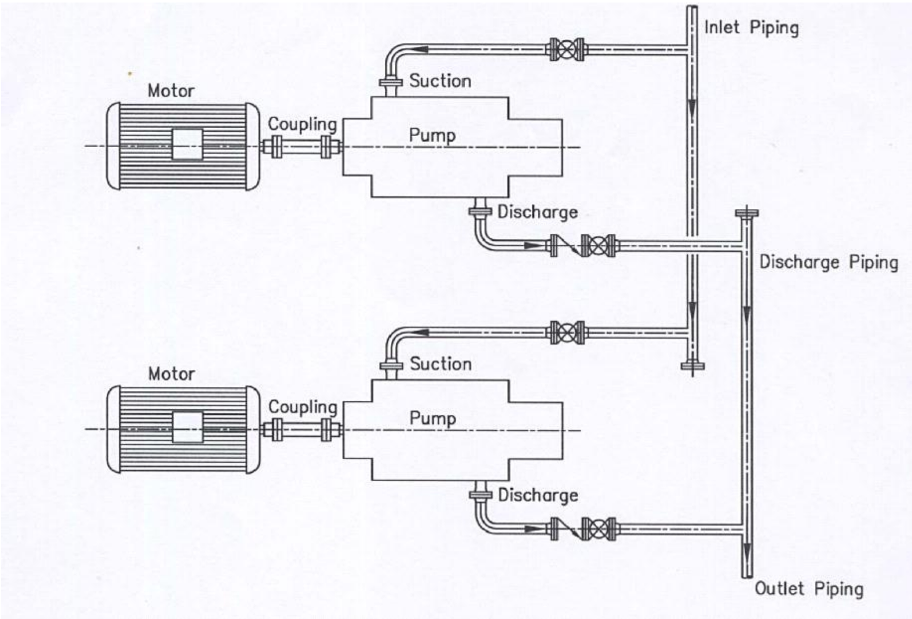


Figure (4.15) Parallel Operation

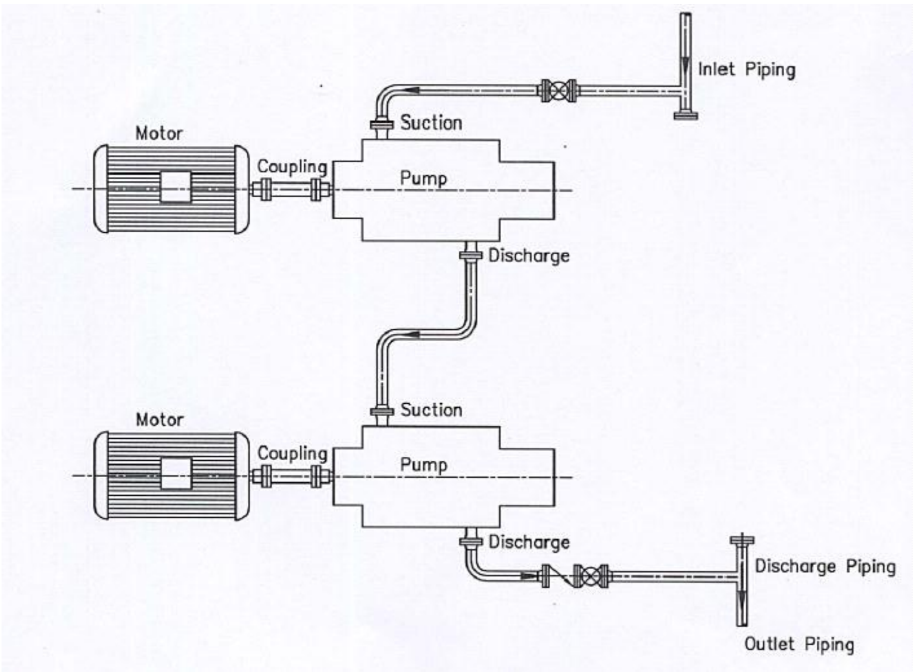


Figure (4.16) Series Operation

4.7.1.1. Pumps in Parallel

When two or more pumps are operated in parallel the combined performance characteristics can be found by adding the capacities at the same head. Thus in the case of two identical pumps the combined characteristics can be found by doubling the capacity at various heads. The capacity at the operating point established by the intersection with the system head curve, however, will be less than twice the discharge obtained from a single pump working on its own in the system.

Where the individual pumps have different characteristics the combined characteristics are obtained by summation of the individual capacities at the same heads. The working points of the individual pumps are again established by the intersection of the system head curve.

It is a general characteristic when centrifugal pumps are operated in parallel that the flatter the head (*H*) and capacity (*Q*) curves of the pumps the more the discharge of the individual pump is reduced.

First consideration is initial condition; the total brake horsepower required is 510 Kw, (255 Kw for each pump) with estimated 77 % pump efficiency and head is 400 m. As shown in figure (4.17), by using two pumps operating in parallel each pump should give (320m) head, (430m^3/h) flow rate, at 74% efficiency, we can operate one pump which will give (180m) head, and (320m^3/h) flow rate.

Performance curves for single pump operation and two pumps operations in parallel are plotted against the system head curves, operating points will at intersections of pump curves and system head curves.

As mentioned above, connecting two pumps in parallel doubles the capacity and the head remains the same, but due to the losses in the system (system head curve), operating two identical pumps in parallel does not necessary give double the capacity because this depends on the system head curve. As shown in figure (4.17) the efficiency is the intersection point between the system head curve and pump curve.

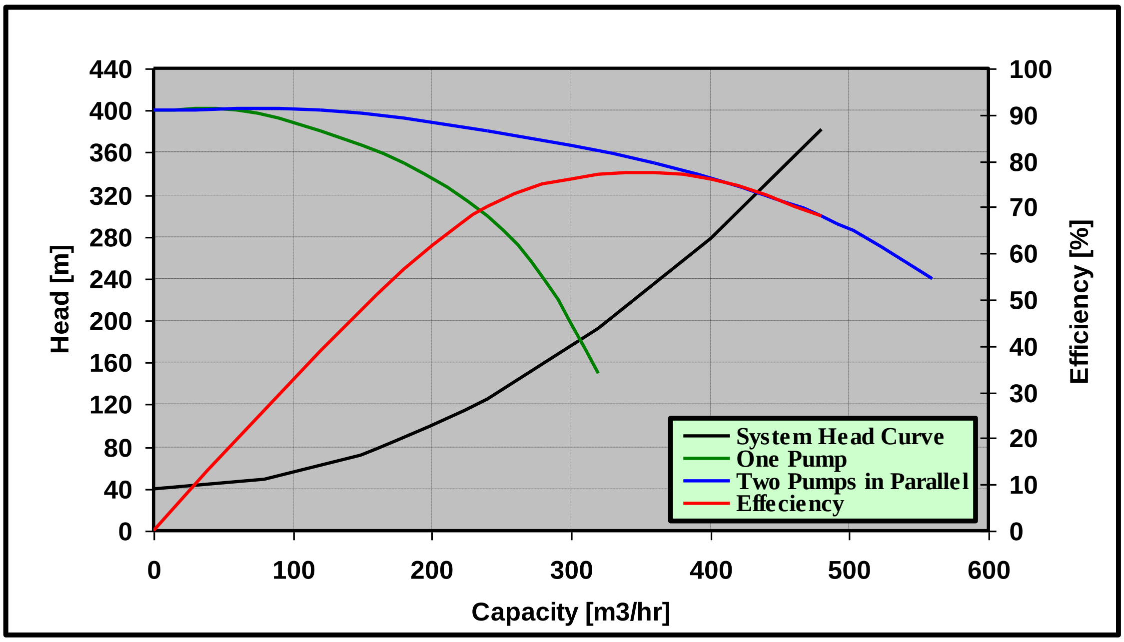


Figure (4.17) Performance of Two Pumps Installed in Parallel against System Curve

4.7.1.2. Pumps in Series

When two pumps are operated in series the capacity will be same for each pump, and the head will be divided into two, for each pump. Thus in the case of two identical centrifugal pumps the combined curve can be plotted by doubling the head ordinates for each value of capacity considered. Where the characteristics are different the respective heads are summed at each capacity point to arrive at the combined characteristics. It follows that the flow rates must be the same, but the working heads on the individual pumps are only the same if the head (*H*) and (*Q*) curves of the individual pumps are identical.

All the centrifugal pumps used by Veba Oil Operations in all its field sites are connected in parallel. No series connections are used.

The pumps should be selected from different offers and manufacturers according to the performance of the pump, initial cost, operating cost, manufacturer history and the purchasing condition.

Chapter 5

COSTS

5.1. Introduction

A detailed estimate of the manufacturing cost only becomes possible when detailed engineering drawing and parts list are available and when quotes have been obtained for all purchased items. Availability of this information implies that development has progressed to an advanced state and that considerable funds have already been spent on development. An early estimate of product cost is highly desirable. A cost estimate based on weight per unit power has the advantage that commercial catalogues usually list the power rating and also the weight of products. Therefore, it becomes relatively easy to establish the current status of commercially available products.

If drawings and parts lists are available, a detailed manufacturing cost estimate can be prepared. Unfortunately, a comparison with competitive products still remains difficult because only their sales price would be known, which includes confidential overhead charges, the amortization of development costs, sales cost, distributor markup, and profit. To get a valid comparison, the cost of competitive product must be evaluated in exactly the same manner as the new product. Drawings and parts list must be prepared for the competitive product.

5.2. Pump Purchasing

In the process of specifying the pumping equipment the engineer is required to determine system requirements, select the pump type, write the pump specification, and develop all information and data necessary to define the required equipment for the supplier.

Having completed this phase of the work, the engineer is then ready to take the necessary steps leading to purchase of the equipment. These steps include issuing the specification, selection of supplier, and release of all necessary data for purchase order issuance.

5.2.1. Evaluation of Bids

Perhaps the most important consideration after having specified the pumps is the manner of evaluating the bids. Evaluation factors that are used should consider not only the first cost but the pump performance, as well as guarantees, economic advantage of alternates, delivery, maintenance, installation service, etc. the sum of these several factors serves to provide the purchaser with a broader base for his purchase decision and will assure more satisfactory performance over the life of the equipment.

It is convenient to separate the evaluation of bids into two categories, those that relate to price, and those that relate to technical features. In this way the technical features which are important to performance but are difficult to quantify are treated separately and can be compared to the cost differences from the priced evaluation.

5.2.1.1. Cost

First cost is a major consideration in the evaluation of bids. In many cases where offerings may be almost identical because of a commonality among equipment supplied by various bidders, the only real difference may be cost. Where cost alone is considered, or where cost is of paramount concern, it is prudent to consider not only the basic price between the first, second, and third bidder, and the percentage of the bid that these differences represent. In many cases it will be found that the difference in bid price may be tenths of percents and thus the absolute values of the difference when viewed in this perspective become less significant. Where this is the case, more detailed attention should be given to the technical factors. [5]

5.2.1.2. Efficiency

Evaluations should include an estimate of the effect of the different efficiencies quoted. This evaluation is normally made at the warranted point (usually full load) while efficiency at other points are treated qualitatively unless extended periods of part-load operations are contemplated. The efficiency should be rated from base, the highest efficiency quoted, with an increasing evaluation penalty against pumps having decreasing efficiency. The actual evaluation or quantification of the effect of the efficiency can be done as shown in the equation.

Power per pumping station = Flow, *BPOD* × $\dfrac{H_T,\ ft \times sg}{136{,}000 \times eff_{motor} \times eff_{pump}}$

$$\frac{45{,}000 \times 946 \times 0.823}{136{,}000 \times 0.83 \times 0.77} \times 0.746 = 301 \text{ Kw}$$

Total power hours per year (5110 hours), two pumps are operational.

$2 \times 5110 \times 1 \times 301 = 3{,}076{,}220$ Kwh

Annual power cost at 0.07 cent per Kwh = $ 215,335 per year

From a similar calculation, but using a pump efficiency of 75 % instead of 77 %, the annual power cost would amount to $ 221,059 an increase of $ 5,724 per year.

5.2.1.3. Economic Life

The economic life of the equipment is a consideration in evaluation of any of the proposals but it is an extremely difficult item to measure. To some extent the weight of the equipment is an indicator of the ruggedness and the durability of the equipment, all other things being equal. Another measure of the economic life would be the speed of the equipment. Thus, if one pump operates at 1500 rpm while an alternative pump operates at 3000 rpm, assuming other construction details are roughly equivalent, it is likely that the 1500 rpm pump, because of its lower speed, will have a longer economic life and will be less likely to require premature replacement.

The concept economic life is extremely difficult to quantify and this type of item is best left to the technical spread sheet where it is dealt with qualitatively. [5]

5.2.1.4. Shipping Cost

Shipping cost may be a factor in the evaluation. The cost for shipping should be determined for each of the alternates, particularly if the pumps are large and require special handling.

The shipping cost also becomes a factor in the assumption of responsibility for any damage to the pump prior arrival at the point of use.

5.2.1.5. Delivery Time

An important element of the quotation is the delivery time required for the pump. The amount of time required to design and manufacturer the pump, ready for delivery, after approval to manufacturer, will vary between bidders but will be generally in accordance with the list shown in table (5.1). [3]

Type of application	Time required
Per-engineered and conventional pumps 6 inch (152mm) discharge and smaller	Off the shelf to 16 weeks
Pumps 6 to 48 inch (152mm to 1219mm) discharge with other than electric motor drivers	12 to 48 weeks
Large pumps	26 to 78 weeks
Special pumps	Negotiated with the manufacture

Table (5.1) Delivery Time [3]

5.3. Operating Costs

The operation costs are annual costs, excluding maintenance and repair costs. Pump station costs depend on many factors including:

- Flow and quantity of the liquid being pumped.
- Depth of the required structures.
- Alternative for standby power sources.
- Operation and maintenance needs and support.

These factors must be examined and incorporated in the preparation of pump station cost estimate.[52]

5.3.1. Maintenance Costs

Maintenance costs are extremely difficult to quantify. It is possible however, using the estimated spare parts to evaluate in a vary rough way what the spare parts costs would be for over a year and to make an estimate of the number of hours required to install at least some of the principal spares. For instance, if the design quoted had two packed glands it might be reasonable to assume that on a yearly basis the two glands should be removed and the packing replaced. If an alternate for a mechanical seal is offered it should be reasonable to expect that a mechanical seal would have a life of perhaps three to five years in clear-fluid service, and the cost, both parts and labours, evaluated on this basis.

5.3.2. Repair costs

Repair costs are unanticipated expenditures that are required to prolong the life of a system without replacing the system.

Some maintenance costs are incurred annual and others less frequently. Repair costs are by definition unforeseen so it is impossible to predict when they will occur. For simplicity, maintenance and repair costs should be treated as annual costs. All maintenance and repair costs are to be discounted to their present value.

It is important to note that all options are not created equal. At first glance, maintenance and repair costs could be judged to be equal for all alternatives.[51]

5.3.3. Spare Parts Costs

It is important to evaluate the cost of spare parts for each of the pumps offered by the bidders. Since the manufacturer is usually in a better position to evaluate the spare parts requirements, it is usual that the specification will call for the manufacturer to provide a priced list of the spare parts which are recommended for one year's maintenance and operation of the equipment. The cost of the spare parts should be reviewed, and if it appears that a bidder's spare parts requirements are either extremely high or low, supplementary information or clarification should be obtained. Either a prebid or pre-award conference with the bidders can be helpful to evaluate these requirements. [3]

During system planning it may be necessary to estimate the cost of system maintenance. The cost of maintenance per year should be in the region of (5-10) per cent of the capital cost, these are total costs which allow for the costs of spares and labour costs of repairmen.[47]

5.4. Cost Calculations

Recurrent costs are considered to consist of two parts: maintenance costs and costs of operation. The annual cost calculations of M. O. L. centrifugal pumps used at Ghani field can be calculated as follows:

From table (4.1) the powers of M. O. L. pump are 255 Kw. And as two out of five pumps are operational, the powers per pumping station are 510 Kw (255 Kw × 2).

Power cost per hour = number of operating hours × power per pumping station × cost of Kwh.

$$= 5110 \times 510 \times 0.07 = \$182,427$$

Also the maintenance cost can be considered as follows:

Cost of one pump = 150,000 + 10% Capital Cost

$$= 150,000 + 15000 = \$165,000$$

So the cost of five pumps is $825,000 (165,000 × 5 Pumps).

The total labours cost per month is $6,000 therefore, the total labours cost per year is $72,000. Table (5.2) shows the annual cost calculation of M. O. L. centrifugal pumps used at Ghani oil field.

M.O.L pumps at Ghani field	Maintenance Costs per year $	Operating Costs per year $		Total Annual Costs per year $
		Labours	Power	
	825,000	72,000	182,427	1,079,427

Table (5.2) Annual Cost of M. O. L. Centrifugal Pumps at Ghani oil Field

A new proposed pumping system can be installed at Ghani oil field with less cost and less number of pumps. Assuming four new centrifugal pumps to be installed at Ghani oil field with 200 Kw each and are connected in parallel. If two out of four pumps are operational, the powers per pumping station are 400 Kw (200 Kw × 2).

From similar calculations, but using pump that costs $160,000 as shown in table (5.3), the total annual costs for the four proposed pumps would amount to $919.080 a decrease of $160,350 per year.

New proposed pumps at Ghani field	Maintenance Costs per year $	Operating Costs per year $		Total Annual Costs per year $
		Labours	Power	
	704,000	72000	143,080	919,080

Table (5.3) Annual Cost of New Proposed Centrifugal Pumps

From table (5.2) and (5.3), it is clear that the total annual costs per year for the new system with four pumps is less than the total annual costs per year for the existing system at Ghani oil field (five centrifugal pumps) by almost 15%.

Chapter 6

RELIABILITY AND AVAILABILITY SYSTEMS

6.1. Introduction

Reliability is the capability of equipment not to break down in operation. When equipment works well, and works whenever called upon to do the job for which it was designed, such equipment is said to be reliable. Satisfactory performance without breakdowns while in use and readiness to perform at the desired time are the criteria of equipment's reliability.

The measure of equipment's reliability is the frequency at which failures occur in time. If there are no failures, the equipment is one hundred per cent reliable; if the failure frequency is very low, the equipment's reliability is usually still acceptable; if the failure frequency is high the equipment is unreliable.

A well designed, well engineered, thoroughly tested, and properly maintained equipment should never fail in operation. However, experience shows that even the best design, manufacturing, and maintenance efforts do not completely eliminate the occurrence of failures.

The pumping system in Ghani oil field is setup in parallel, if one component or pump fails, then the systems are expected to be working normally, to be active, and each one is capable of meeting the functional requirements placed on the overall system.

The reliability of the system will depend crucially on the reliability of the components present. This chapter shows how to calculate system reliability from individual component reliability and gives a number of practical systems.

6.2. Reliability of Series Systems

Figure (6.1) shows the components consisting of the system with individualities reliability R_1, R_2, R_3, R_4, R_5 respectively. The system will only survive if every element survives; if one element fail then the system fails. The reliability of each element is independent of the reliability of the other elements.

The probability that the system survives is the probability that component one survives and the probability that two survives and the probability that three survives and the probability that four survives and that the probability that five survives. The total reliability will decrease, the system reliability R $_{(SYST)}$ is therefore:

$$R_{(STS)} = R_1 \times R_2 \times R_3 \times R_4 \times R_5 \tag{6.1}$$

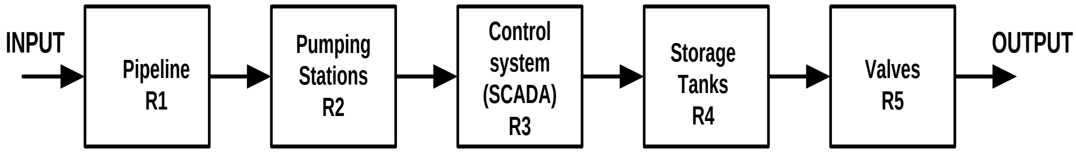


Figure (6.1) Reliability of series components

In the system each component can be in described by a constant failure rate λ. The reliability of each component is given by the following equation: [46]

$$R = e^{-\lambda t} \tag{6.2}$$

Thus

$$R_{(SYS)} = e^{-(\lambda_1 + \lambda_2 + \lambda_3 + \lambda_4 + \lambda_5)t} \tag{6.3}$$

Failure rate of the system = $\lambda_1 + \lambda_2 + \lambda_3 + \lambda_4 + \lambda_5$ (6.4)

Where:

R = Reliability, (probability of no failure in time t)

e = Base of the natural system of logarithms

λ = Constant failure rate

$R_{(SYS)}$ = Reliability of the system

As *MTBF* (Mean Time Between Failure) = $\frac{1}{\lambda}$ (6.5)

Or $\lambda = \frac{1}{MTBF}$ (6.6)

Then Reliability = $e^{-t/m}$ (6.7)

Where:

m = MTBF

t = Time of the Reliability

6.3. Reliability of Parallel Systems

Figure (6.2) shows an overall system the component consisting of the pumping system individual elements or systems in parallel with individual unreliability F_1, F_2, F_3, F_4, F_5 respectively. All the component or systems are expected to be working normally, i.e. to be active, and each one is capable of meeting the functional requirements placed on the overall system. However, only one component/system is necessary to meet these requirements; the remainder increase the reliability of the overall system.

The overall system will only fail if every component/system fails; if one component/system survives the overall system survives.

The reliability of each component/system is independent of the reliability of the other component, then the probability that the overall system fails the probability that component/system one fails and the probability that two fails and the probability that three fails and the probability that four fails and the probability that five fails. The overall system unreliability $F_{(SYST)}$ is therefore: [46]

$$F_{(SYS)} = F_1 \times F_2 \times F_3 \times F_4 \times F_5 \qquad (6.8)$$

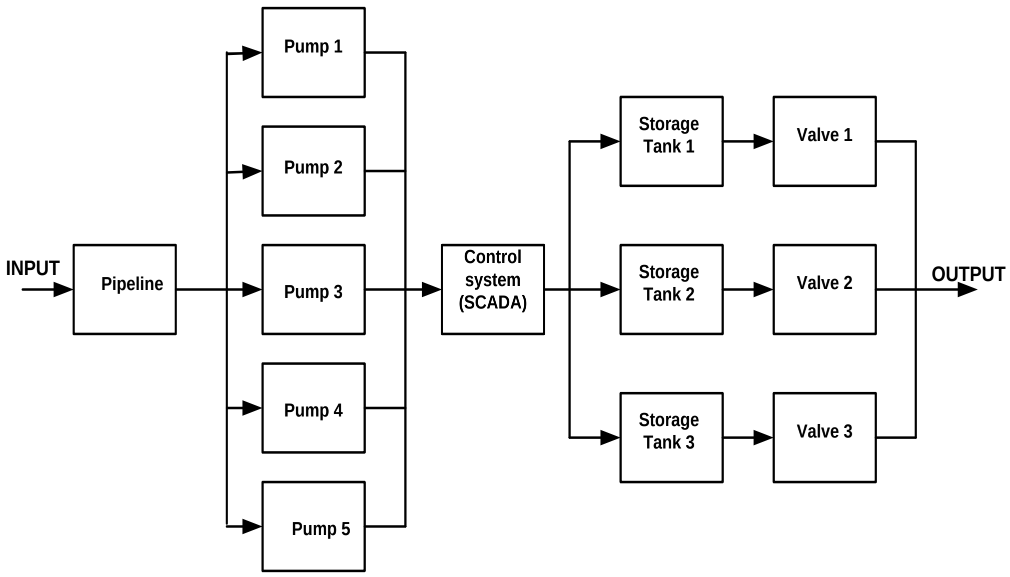


Figure (6.2) Reliability of Parallel Components

Comparing equations (6.1) and (6.8), we see that for series systems, system reliability is the product of component reliabilities, whereas for parallel systems, system unreliability is the product of component unreliability.

6.3.1. Standby Redundancy

The standby redundant system configuration is form of paralleling where only one subsystem is in operation; brought into operation.

In some critical systems, like Ghani pumping system it is essential to have one or more standby units to substitute the failure of the main units, figure (6.3) shows five identical pumps, each with failure rate λ and a switching system (S). Normally two pumps are operating and the other pumps are shut down. If pump one or two fails then pump three is switched in, if pump three fails then pump four switched in, if another pump fails then pump five is switched in. Assuming the switching system has perfect reliability (R_S = 1), then the reliability of the standby system can be given by the cumulative poison distribution. [48]

$$R\ (t) = \exp(-\lambda\, t) \sum_{k=0}^{n=1} \frac{(\lambda t_R)^K}{K} \quad \text{(Reliability of standby system)} \qquad (6.9)$$

Thus for n = 1, we have as expected:

$$R_{(t)} = \exp(-\lambda\, t) \qquad (6.10)$$

Thus for n = 2, R (t) is increased to:

$$R\ (t) = \exp(-\lambda\, t)\ [1+\lambda\, t] \qquad (6.11)$$

The term $\lambda\, te^{-\lambda t}$ represents the increases in reliability due to adding one standby pump. For n = 3, $R_{(t)}$ is further increased to:

$$R\ (t) = \exp(-\lambda\, t) \left[1+\lambda t+\frac{1}{2}(\lambda t)^2\right] \qquad (6.12)$$

The term represents the further increase in reliability due adding a second standby pump unit.

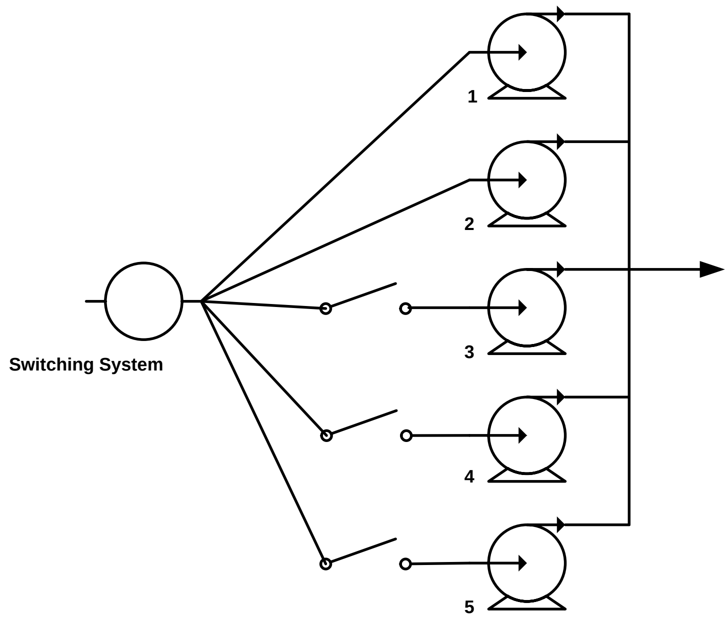


Figure (6.3) Standby System

6.3.2. Failure Rate Data

Table (6.1) and figure (6.4) show some observed average Mean Time Between Failure (MTBF) and their failure rate for the main components in the system that is used by Veab Oil Operations; this data has been provided by operations and maintenance department at Ghani oil field.

Components	MTBF (year)	Failure Rate (year) λ
Pipeline (130 Km) from Ghani to Joffra	25	0.04
Centrifugal Pumps + Driver	6	0.166
Control System	18	0.05
Storage Tanks	27	0.03
Valves	22	0.045

Table (6.1) MTBF for Different Components

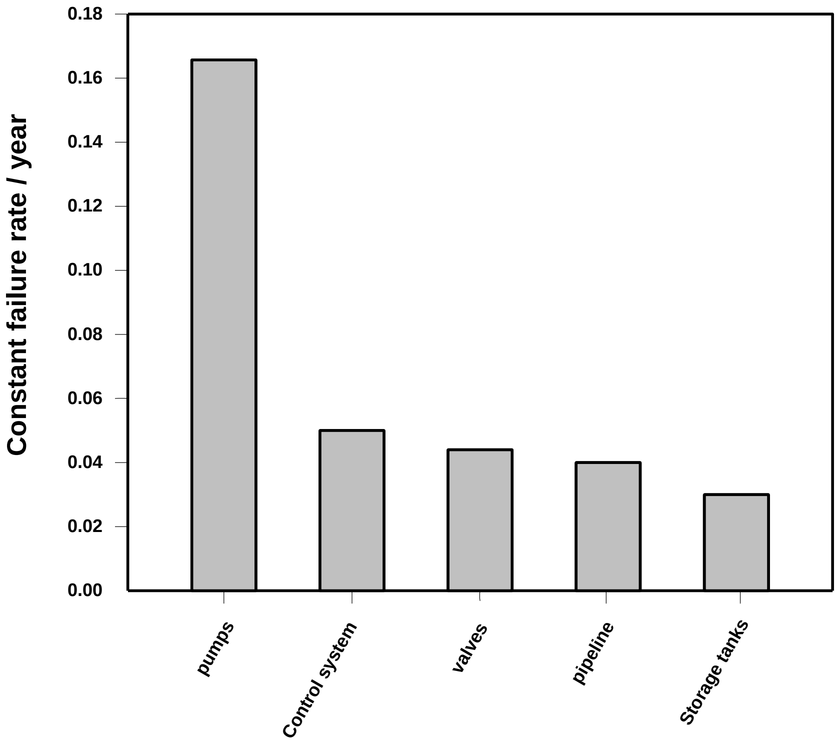


Figure (6.4) Failure Rate of the System Components

In general the failure rate of a component or element depends on four main factors:

- Quality
- Temperature
- Environment
- Stress

6.4. Reliability Calculations

6.4.1. Reliability of the Pumping Systems

1) Reliability of one pump = $e^{-\lambda t} = e^{-(0.166)1 \times 1} = 0.847$

   Unreliability $F = 1 - R = 0.153$

   F^5 (Probability of none working) = $(0.153)^5 = 0.00008$

   $R = 1 - F$

Total probability of at least one working = 1- 0.00008 = 0.99992

6.4.2. Reliability of the Systems

As mentioned in reliability of parallel systems above the system in Ghani oil filed is a parallel so the total reliability of the system is given by:

$R_{(SYSTEM)} = R_{(pipeline)} \times R_{(pumping\ station)} \times R_{(control\ system)} \times R_{(storage\ tank)} \times R_{(valves)}$

$R_{(pipeline)} = e^{-\lambda t} = e^{-(0.04) \times 1} = 0.970$

$R_{(pumping\ station)} = 0.99992$

$R_{(control\ system)} = e^{-\lambda t} = e^{-(0.05) \times 1} = 0.951$

$R_{(sub.\ system)} = R_{(storage\ tank)} \times R_{(valves)}$

$R_{\text{(storage tank)}} = e^{-\lambda t} = e^{-(0.03)\times 1} = 0.970$

$R_{\text{(valves)}} = e^{-\lambda t} = e^{-(0.045)\times 1} = 0.956$

$R_{(SS)} = R_{(ST1)} \times R_{(V1)} = 0.970 \times 0.956 = 0.927$

$R_{(SS)} = R_{(ST2)} \times R_{(V2)} = 0.970 \times 0.956 = 0.927$

$R_{(SS)} = R_{(ST3)} \times R_{(V3)} = 0.970 \times 0.956 = 0.927$

$R_{\text{(sub. system)}} = 1 - (1 - 0.927)^3 = 0.99961$

$R_{(SYSTEM)} = R_{\text{(pipeline)}} \times R_{\text{(pumping station)}} \times R_{\text{(control system)}} \times R_{\text{(sub. system)}}$

$= 0.970 \times 0.99992 \times 0.951 \times 0.99961 = 0.922$

6.5. Availability of the Pumps

When a repairable product is up, i.e. working satisfactorily, it is available for use. When the product is down, i.e. being repaired, it is unavailable for use. It is important to have an average measure of the degree to which the product is either available or unavailable. The availability (A) and unavailability (U) can be calculated using the following: equation.

$$\text{Availability} = \frac{\textit{Total up time}}{\textit{Total up time} + \textit{Total down time}} \qquad (6.13)$$

$$U = 1 - A \qquad (6.14)$$

Total down time = Down time per pump failure $\times$ Failure rate λ (6.15)

The availability depends on reliability; availability can therefore be increasing by increasing mean time between failures (*MTBF*), i.e. reducing mean failure rate. *A* and *U* depend on mean down time (*MDT*), availability can be increased by reducing (*MDT*). Thus availability also depends on maintainability, i.e. how quickly the product can be repaired and put back into service. [46]

Many engineering systems contain elements or subsystems that can be repaired. However, repair of failed elements that are down can take place simultaneously with the normal operation of working elements that are up.

6.5.1. Availability Calculations

Time working per day = 14 hours $\times$ 365 days = 5110 hours per year.

Time shut off per day = 10 hours $\times$ 365 days = 3650 hours per year.

Down time per pump failure = 14 hour $\times$ 14 days = 196 hours.

From equation (6.15), Total down time = 32.536

$$\text{Availability} = \frac{Total\ up\ time}{Total\ up\ time + Total\ down\ time}$$

$$= \frac{5110}{5110 + 32.536} = 0.9936$$

Therefore availability of one pump = 0.9936

From equation (6.14) Unavailability (U) = 0.0064

The availability time, which is equal to the fraction of time (P) for which the minimum requirement of power is met, can be calculated by using the following equation: [53]

$$P = \frac{N!}{K!(N-K)!} \times (1-A)^K \times A^{N-K} \qquad (6.16)$$

Where:

P = Fraction of Time

N = Total Number of Pumps

K = Number of Pumps Shut Down

A = Availability

Cases	Availability	Working Hours	Total Pumps
Case 1	0.9936	5110	3
Case 2	0.9936	5110	4
Case 3	0.9936	5110	5
(Case 1) Availability of Two Pumps Working and One Standby			
Number of Pumps Installed	Pumps Shut Down	"P"- Shut Down Time	Availability
(N)	(K)	(P)	(A)
3	0	0.981	0.9936
3	1	0.018	0.9936
	Availability Time	0.999	
(Case 2) Availability of Two Pumps Working and Two Standby			
Number of Pumps Installed	Pumps Shut Down	"P"- Shut Down Time	Availability
(N)	(K)	(P)	(A)
4	0	0.974	0.9936
4	1	0.0251	0.9936
4	2	0.0002	0.9936
	Availability Time	0.9993	
(Case 3) Availability of Two Pumps Working and Three Standby			
Number of Pumps Installed	Pumps Shut Down	"P"- Shut Down Time	Availability
(N)	(K)	(P)	(A)
5	0	0.9684	0.9936
5	1	0.0310	0.9936
5	2	0.0004	0.9936
5	3	0.258×10^{-5}	0.9936
	Availability Time	0.9998	

Table (6.2) Availability of Pumping Systems

Chapter 7

CONCLUSION AND RECOMMENDATIONS

7.1. Conclusion

As discussed earlier crude oil is piped from Ghani oil field to Joffra oil field then to Ras Lanuf Terminal for a total distance of about 165.6 miles (265Km). This crude oil is transported from Ghani oil field to Joffra oil field by the use of five centrifugal pumps installed in Ghani oil field. Usually two out of five centrifugal pumps are operational the other three pumps are on standby in case of failure.

This study has considered the operation of these centrifugal pumps to arrive at an optimum pumping system design with least cost and highest reliability.
In pumping system design the optimum design should be selected from different options. So this study has proposed different centrifugal pumps and their impact on the whole system from Ghani oil field to Joffra oil field and then analysis the cost of the proposed design.

The main change will be in pumping station design mainly to decrease the number of existing centrifugal pumps with less cost and to meet Veba Oil Operations (VOO) requirements. This may be achieved by installing fewer numbers of pumps and connecting them in parallel, thus giving the same head.

The annual system costs for the existing centrifugal pumps used at Ghani field and the new proposed pumps to be used at Ghani field have been discussed and calculated. The cost calculations show that the total annual costs for the new proposed system is less than the annual costs for the existing system used at Ghani field by almost 15%.

The reliability and the availability of the selected system have been discussed in chapter 6. The calculations of the reliability and the availability have been calculated for the existing five centrifugal pumps installed at Ghani oil field and the four proposed pumps to be installed at Ghani oil field. The calculations show that the reliability and the availability of existing and the proposed centrifugal pumps are nearly the same.

7.2. Recommendations

- It is necessary to perform economic calculations to compare the pumping system design with other combinations of efficiency, power, speed, availability and reliability in order to choose the best system.

- Reliability and availability of the system should be considered to make sure the system is at acceptable level of reliability and availability.

- Replacement of the existing pumps at Ghani oil field with the proposed pumps will save about 15 % of the total annual cost of the existing centrifugal pumps. See tables (5.2 and 5.3).

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